JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS.

VOL. IX. MAY, 1897. No. 2.

The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY
(Under whose supervision this number is published)
Chief Engineer DAVID SMITH, U. S. Navy.
Chief Eng'r A. B. CANAGA, U. S. N. Passed Ass't Eng'r H. P. NORTON, U. S. N.
Passed Ass't Eng'r F. C. BIEG, U. S. N. Passed Ass't Eng'r EMIL THEISS, U. S. N.

THE CONSTRUCTION, INSTALLATION AND PERFORMANCE OF THE ELECTRIC LIGHTING PLANT AT THE GOVERNMENT HOSPITAL FOR THE INSANE AT WASHINGTON, D. C.

BY CHIEF ENGINEER G. W. BAIRD, U. S. NAVY.

The following description of the electric lighting plant at the Government Hospital for the Insane, and the data of its performance, are from my report to the Honorable the Secretary of the Interior.

The plant is in a brick building, situated in the grounds of the Hospital, near the wharf, and close to the river side. The circulating water for the condensers is pumped from the river, the height of lift varying from 9 to 12.5 feet under normal conditions, but reaching as much as 17 feet on rare occasions. The water is drawn through a 6-inch pipe, a distance of about 342 feet, the pipe having a foot valve at its lower end.

The specifications for the plant were prepared by myself, and I had general charge of the construction, installation and testing of it, but not of the disbursement of the appropriation. The general plan and leading dimensions of the conductors were given by myself, but the details of the wiring were worked out

by Mr. M. J. O'Donnell, who personally supervised the installation of all the conductors and appliances. This required unusual skill and attention, as many of the buildings are old, necessitating careful insulation; besides which it was essential to place the conductors and appliances out of the reach of the more than 1,700 patients.

The output of the two large dynamos was stipulated to be 100 kilowatts, and of the two small dynamos, 50 kilowatts, with 125 volts E.M.F. at the poles, and a commercial efficiency of 80 per cent. The winding of neither field nor armature was to heat more than 70 degrees Fahr. above the temperature of the room, when running at full load. The large engine was required to develop 200, and the small engine, 100 I.H.P., with 100 pounds pressure of steam at the boilers, 25 inches of vacuum in the condenser, and 265 and 285 revolutions per minute, respectively.

The insulation resistance of the dynamos, conductors and switchboards was specified to be not less than 500,000 ohms.

The contract for the above machinery was, after competitive bidding, placed with the General Electric Company. The boilers, also from the specifications of the writer, were built by the E. Keeler Company of Williamsport, Pa.

BOILERS.

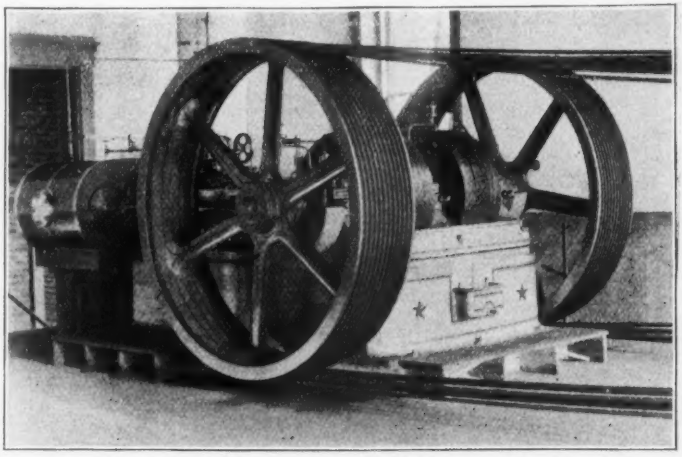
There are two ordinary steel shell boilers, externally fired, set in brick work, debouching their products of combustion into a single smoke pipe. There is a damper in the uptake of each boiler. The tubes are of wrought iron. The principal dimensions are as follows:

Diameter of shells, feet	6
Length of shells, feet	14
Number of tubes in each boiler	122
External diameter of the tubes, inches	3
Thickness of tubes, No. 12 B. W. G., inch	.109
Length of the tubes, feet	14
Spacing of the tubes, vertically and laterally, inches	4
Number of furnaces to each boiler	1
Width and length of the grates, feet	6
Area of grate surface in each boiler, square feet	36
over one bridge wall for draft, square feet	10

Area through tubes of one boiler for draft, square feet	5.15
uptake of one boiler for draft, square feet	4.56
chimney for draft, square feet	8.75
Water heating surface in each boiler, square feet	1,376
Ratio of grate surface to area over the bridge wall	3.6
through the tubes	7
uptake	7.9
the smoke pipe	8.23
water heating surface to grate surface	38.22
Height of chimney above grates, feet and inches	57-6
Extreme length of brick work in each boiler, feet and inches	18-3
width of brick work of both boilers, feet and inches	17-5
height of brick work above floor, feet and inches	10-8
Width of top edge of grate bars, inch	$\frac{5}{8}$
air space between grate bars, inch	$\frac{5}{16}$

ENGINES.

The engines are of the horizontal, direct-acting, two cylinder, compound type, with opposite cranks, and have piston valves, with an automatic cut-off governor affecting only the high pressure valves. There are no steam jackets. The two driving wheels on each engine are scored for $\frac{3}{4}$-inch ropes. Each engine has a Stratton separator, steam trap, independent air pump and



THE LARGE ENGINE.

jet condenser. The engines were built by the Armington & Sims Engine Co.

	Large engine.	*Small engine.*
Diameter of H.P. cylinder, inches	13	10½
H.P. and L.P. piston rods, inches	2¼	1½
Net area of H.P. piston, square inches	130.75	85.71
Displacement of H.P. piston per stroke, cubic feet	1.13	.59
Clearance space and steam passages at one end of H.P. cylinder, cubic feet	.105	.05
Ratio of clearance to piston displacement	9:100	8.4:100
Area of steam port of H.P. cylinder, square inches	5.7	3.6
exhaust port of H.P. cylinder, square inches	14.3	8.5
Clearance of H.P. piston, inch	¼	5/16
Diameter of L.P. cylinder, inches	20½	16½
Net area of L.P. piston, square inches	338 08	212.94
Stroke of pistons, inches	15	12
Displacement of L.P. piston per stroke, cubic feet	2.85	1.48
Space in clearance and steam passages at one end of L.P. cylinder, cubic foot	.22	.15
Ratio of clearance to piston displacement	1:13	1:11
Area of L.P. steam port, square inches	16	10
exhaust port, square inches	35	21
Clearance of L.P. piston, inch	¼	7/32
Number of driving wheels	2	2
Diameter of driving wheels, on pitch line, inches	74	74
Weight of driving pulley, pounds	3,095	2,140
regulator pulley, pounds	3,080	2,105
Number of ropes in each drive	11	6

AIR PUMPS AND CONDENSERS.

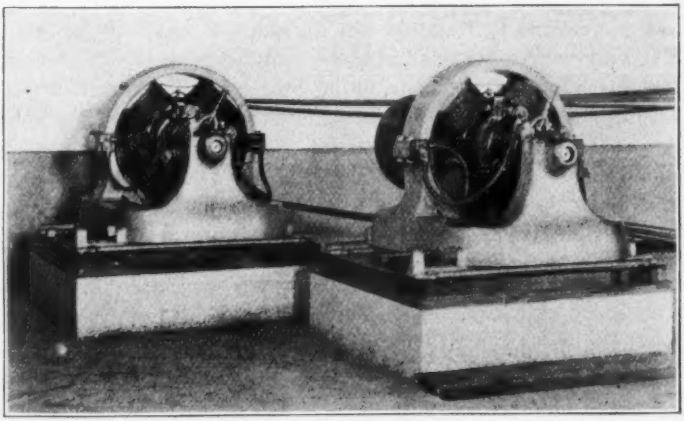
The condensers are cylindrical in form, with hemispherical tops and flat bottoms. The air pumps are horizontal, direct-acting. The air pumps and condensers were built by Mr. M. T. Davidson, Brooklyn.

	Large.	*Small.*
Diameter of condenser, inside, inches	22¾	19⅞
Axial length of condenser, inches	35	32
Volume of condenser, cubic feet	7.25	5.25
Diameter of injection pipe, inches	3½	3
discharge opening, inches	8	6
steam piston of air pump, inches	10	7
pump and piston rod, inches	1¾	1¼
Net area of steam piston, square inches	77.34	37.86
Area of steam port, square inches	1.87	1

	Large.	*Small.*
Area of exhaust port, square inches	3.14	1.22
Clearance of steam piston at each end, inch	$\frac{3}{4}$	$\frac{1}{2}$
per cent	7	6
Volume of clearance at one end, cubic inches	75	27
Diameter of pump piston, inches	12	9
Net area of pump piston, square inches	111.89	63
valves in air pump, square inches	23.5	16.5
Stroke of pistons, inches	14	12
Net displacement of pump piston per stroke, cubic foot	0.91	0.44

FEED PUMP.

A Knowles boiler feed pump, No. 7, having a 10-inch steam piston, 6-inch water piston and a nominal stroke of 12 inches, was provided by the Superintendent of the Hospital. The nominal displacement of the steam piston is .54, and of the water piston, .19 cubic feet. It is much too large for its present purpose; it is intended, however, to be used to feed an additional number of boilers which are to be installed in the future. The slip and loss of action of the pump, at its necessarily low speed of piston, as hereinafter recorded, is great; but as the hydrant pressure exceeds the boiler pressure, that pressure will be utilized for feeding the boilers, and the pump will, probably, seldom be used.



THE 50-KILOWATT DYNAMOS.

DYNAMOS.

Each engine drives two dynamos, which are in series for the three-wire system. The dynamos are multipolar, compound wound and self regulating. Each dynamo has four poles, is wound for 125 volts, and has the usual rheostats, operated from the switch board. The output of each of the larger machines is figured to be 50 kilowatts at 875 revolutions, and of the smaller ones, 30 kilowatts each at 1,000 revolutions per minute.

ROPE DRIVE.

The transmission is effected by ropes which are four-strand manilla, with a center strand laid in tallow. They are ¾-inch in diameter and the splices are 10 feet long, each rope being separate. The grooves are of the same design as made by the writer eight years ago for the extensive rope drives in the gun shops at the Washington Navy Yard. Each large dynamo is driven by eleven, and each small one by six ropes.

SWITCH BOARDS.

There are two switch boards, one at the power house, called the station switch board, the other near the "center building," and called the distributing switch board. On the former are two volt meters and four ampere meters, with bus bars, switches, etc. The volt meters are connected by switches with the machines, the bus bars and the distributing switch board, to enable the operator at the station to ascertain the voltage at either point. The boards are of polished slate, with all brass ornamental mountings. The mains, from the machines to the switch boards, run under the floor, through terra cotta conduits, and are thus out of sight. The "outgoing" switches from the station board are evenly divided into two parts.

MAINS.

The main conductors, 11 in number, between the two switch boards, are of hard copper and are carried on high poles. Four of these conductors, aggregating 846,400 circular mils, are positive, four of same size are negative and three are neutral.

It is intended that these mains shall be doubled when the funds permit. The length of the conductor is about 2,200 feet. Besides the main conductors, the poles carry three pressure and two telephone wires.

On the distributing switch board there are two receiving and nine distributing switches. From this board to the various buildings, the conductors are heavily insulated with Kerite, and encased in wooden mouldings; they are run through tunnels, which are from 30 to 36 inches wide and from 6.5 to 7.5 feet high in the clear. All the tunnels are quite dry excepting one; in the latter, for a space of about 400 feet where moisture appears, lead covered conductors were used.

The drop or loss of voltage between the power house and the buildings was calculated to be 10 per cent. when all the conductors are in place and the total installation of 3,000 lamps in circuit. Owing to the limited appropriation, only half the intended number of mains has been placed on the poles. When glowing 2,000 lamps, we found the drop to be 12 per cent. between the station and the building, and with 1,000 lamps, between 6 and 7 per cent. Between the distributing switch board and the lamps, the drop was less than one volt.

FEEDERS AND BRANCHES.

The feeders and branches are, as far as possible, "concealed"; they are run in the conventional way through the joists under the floor, porcelain insulators being used invariably to keep the already insulated conductors from contact with the wood. All splices, taps or joints, where soldered, are re-insulated, after being washed with a solution of carbonate of soda to neutralize any free acid that might be left from the muriate of zinc used in soldering. Whenever conductors were "fished" through walls, behind plastering or around corners, they were covered with circular loom tubing. The feeder and branch switches in the buildings are placed in niches in the walls, against which hinged and locked doors are placed, precluding the possibility of patients reaching them.

APPLIANCES.

All the cut out blocks, junction boxes, switches, etc., are on porcelain bases, and were made by the General Electric Company according to the Edison patent. The lamp sockets and the large knife switches are also Edison's; the small switches are of the Paiste make. The cut outs are either triple or double pole.

MANNER OF MAKING THE TEST.

The scales for weighing the feed water were previously compared and found to be correct. The coal scales were new, and by comparison with a known weight, were also found to be correct. The pressure gauges on the boilers were tested and the corrections made. The Tabor indicators were borrowed from the Bureau of Steam Engineering for the test; the springs were tested in the New York Navy Yard, and corrections applied. The volt meters on the switch boards agreed exactly with the writer's portable meter, which latter had been recently verified by the makers. The thermometers were commercial instruments and were assumed to be correct. The coal and water were weighed. The coal was Cumberland, of good quality, from the mine of "The Consolidated Coal Company" of Maryland. The analysis reported by the contractors is as follows:

Moisture,	.260
Volatile matter,	13.280
Fixed carbon,	81.953
Sulphur,	.077
Phosphorus,	.015
Ash,	5.060
Total,	100.645

The feed water was obtained from the artesian wells near the power house, and is cleaner than the river water. It was heated in the pump house near by, before its delivery to the feed cask. It was weighed in a cask and delivered to a feed tank, from which it was drawn by the feed pump. The height of water in this tank and in the boiler gauge glasses was marked at the beginning of

the test, and brought to the same level at its termination. The temperature of the feed was measured in the feed tank.

During the test of the small plant, I discovered that the feed tank had been permitted to overflow several times, losing some of the water which had been weighed. For this reason I rejected the record of feed water for that test. In the test of the other plant, a much larger feed tank was used, and, from careful observation, I am sure a correct record was kept.

In the test of the smaller plant, one boiler was used; in that of the larger plant, both boilers were used. The rate of combustion, temperature of feed, quality of coal, etc., being practically the same in both tests, the evaporative efficiency is assumed to be the same.

The feed pump exhausted into the atmosphere, and it was observed that there was a continuous escape of steam from the exhaust pipe, due to leaky valve or piston, of which no account could be taken. The quantity of steam used by this pump is calculated from the terminal pressures in the steam cylinder, and the observed piston displacement.

The coal was weighed in barrows, 250 pounds at a charge. The firemen were instructed to regulate the rate of combustion by use of the uptake dampers and ash pit doors, maintaining the pressure at about 95 pounds, to keep the grates covered and the fires heavier in front. The ashes were weighed dry. The fires were shaken out and the ash pits hauled at the beginning of the tests, and the quantity of coal in the furnaces estimated. The fires were again shaken out at the termination of the tests and were brought to the same thickness, as nearly as could be estimated, and all the ashes remaining in the pits were held as coming from the coal consumed during the test.

The speed of the engines was obtained by hand speed gauges, of two different makes, and by a portable tachometer as a check, all correct, I believe, to within a fraction of a revolution. The strokes of the feed pumps were taken from a counter on the pump. The revolutions of the engines and dynamos, steam pressure, vacuum, electro motive force, current, the strokes of air pumps and the temperatures were noted every 15 minutes, and

the averages for the hour recorded. The temperatures of the poles were recorded hourly from a thermometer which was lying on a pole piece. At the termination of a test, and immediately after the dynamos were stopped, the thermometer was inserted into the air spaces of the armatures and the temperature noted.

The steam required to operate the air pumps is calculated from the terminal cylinder pressures, obtained by an indicator. The number of amperes of current in the table is the sum of the readings on the two ammeters; the electro motive force is the mean of the readings of the two volt meters.

The conduct of the test was under my own supervision, and I was ably assisted by Passed Assistant Engineer Emil Theiss, U. S. N. All indicator diagrams were taken by Mr. Theiss or myself and all records kept by one of us. I am also indebted to this officer for valuable assistance in working up the data.

TABLE CONTAINING THE DATA AND RESULTS OF THE TEST OF THE ELECTRIC LIGHTING PLANT OF THE GOVERNMENT HOSPITAL FOR THE INSANE AT WASHINGTON, D. C.

	Small plant.	*Large plant.*
Date of making the test	19 Nov., '96	1 Dec., '96
Duration of test in hours	12	12
Total pounds of coal consumed	3,000	5,882
ashes	245	575
combustible consumed	2,755	5,307
feed water pumped into the boilers	...	53,785
strokes of the feed pumps	4,176	6,216
MEAN QUANTITIES.		
Steam pressure in boilers, in pounds per square inch above atmosphere	93.41	92.33
Fraction of stroke of H.P. piston completed when the steam was cut off	0.33	0 41
Number of times the steam was expanded	7.52	6.30
Vacuum in the condensers, in inches of mercury	23.6	25.2
Revolutions of the engines per minute	287.33	252.54
Pounds of coal consumed per hour	250	490.17
Pounds of feed water pumped into the boilers per hour	...	4,481.7
Current generated by dynamos, in amperes, per hour	485.5	927.6
Electromotive force, in volts	117.8	117.4
Number of kilowatts per hour	57.2	108.9
Number of strokes of air pump per minute	143	86.25
Stroke of the feed pump per minute	5.8	8.24
Actual length of stroke of feed pump in inches	$11\frac{4}{8}$	$11\frac{7}{16}$

MEAN TEMPERATURES, IN DEGREES FAHRENHEIT.

	Small plant.	*Large plant.*
Atmosphere	61 7	31.3
Fire room	68.4	58
Engine room	75·9	74
Refrigerating water admitted to the condenser	54	37.6
Water discharged from the condenser	97.9	101.1
Feed water	129 9	125.9
Poles of the dynamos	103.6	95·5
Armature, at end of test	125	126
Increase in temperature of dynamo fields above engine room,	27.7	21.5

HORSE POWER.

	Small plant.	*Large plant.*
I.H.P. required to work the engine, rope drive, and to revolve the unloaded armature in the magnetic field	16 54	22.56
I.H.P. developed in the small cylinder, loaded	52.76	99.72
large cylinder, loaded	50.88	97.35
Aggregate I.H.P. developed by the engine, loaded	103.64	197.07
Net horse power applied to the revolution of the loaded armature	87.10	174.51
Electrical horse power developed in the pair of dynamos	76.66	145.98
I.H.P. developed by the air pump steam cylinder	1.52	2.19
feed pump steam cylinder	0.48	0.69
Percentage of the net horse power applied to the revolution of the armature in the magnetic field, which appeared in electrical horse power at the poles	88.00	83.65
Aggregate I.H.P. developed by the engines and pumps	105.64	199.96
Mean number of pounds of coal per I.H.P. per hour for each plant	2.37	2.45
Mean number of pounds of feed water per I.H.P. per hour for each plant	21.63	22.42

VAPORIZATION.

	Both boilers.
Mean number of pounds of water vaporized per hour	4,480.8
Mean number of pounds of water vaporized per hour for one pound of coal	9.14
Number of pounds of water that would have been vaporized per pound of coal had the feed been supplied at 100 degrees	8.93
Number of pounds of water that would have been vaporized per pound of coal had the feed been supplied at 212 degrees	9.93
Pounds of water vaporized for one pound of combustible	10.13
Pounds of water that would have been vaporized per pound of combustible had the feed been supplied at 100 degrees,	9.90
Pounds of water that would have been vaporized per pound of combustible had the feed been supplied at 212 degrees,	11.23

INSULATION RESISTANCES.

The insulation resistances between the dynamo leads and the ground, and between the lead and ground as measured by a Wheatstone bridge, were, in ohms:

No. 1 dynamo.	Between dynamo leads and ground	2,000,000
	with leads connected and ground	1,000,000
No. 2 dynamo.	Between dynamo leads and ground	1,000,000
	with leads connected and ground	1,000,000
No. 3 dynamo.	Between dynamo leads and ground	2,000,000
	with leads connected and ground	2,000,000
No. 4 dynamo.	Between dynamo and leads	1,000,000
	with leads connected and ground	1,000,000
The insulation across the line to the ground was		500,000

FULFILLMENT OF CONTRACT.

Boilers.—The boilers were designed, each, to equal a commercial 100 horse power boiler, *i. e.* to vaporize 30 pounds of water per horse power per hour. On the trial, with the low rate of combustion of 6.8 pounds of coal per square foot of grate, the two boilers supplied enough steam to develop 199.96 I.H.P. There was no indication of priming; no appearance of leak nor any mishap whatever, during the two tests. The boilers are abundantly large for the purpose.

Engines.—The engines developed the required power with a less pressure than guaranteed; they regulated within 1½ per cent. between full load and no load, though considerable adjustment was necessary to the governor of the large engine before it came within 1½ per cent. The engines ran without thump or jar, and without heating.

Air pumps and condensers.—The air pumps and condensers worked well during the entire tests, keeping the vacuum from 23.5 to 26.5 inches and condensing the steam easily.

Rope drives.—The rope drive ran very quietly, with very little slip, and gave almost identically the same speed to the respective pairs of dynamos. During the test of the large plant, the engine room became warm at one time, causing the tallow in the center cores of the new ropes to flow, which occasioned considerable

slip; but, on opening the windows and cooling the room, the ropes gripped again.

Dynamos.—The large pair of dynamos exceeded the requirements by 9.27 per cent., and the small pair, by 7.19 per cent. They required the usual amount of adjusting and regulating, and were run several hours unloaded, to dry the armatures well before they were allowed to be excited. During the tests they ran without over-heating or sparking. The voltage could not be run up to 125, as the lamps began to "boom" before that pressure was reached. The mean loads of 108.9 and 57.19 kilowatts on the large and small plants, respectively, were obtained by increasing the amperage, which was a severer test of the machines. A mistake had been made in speeding the large engine by about 13 revolutions less than required, which I did not discover until the test had commenced. Finding that the load was so easily carried, as above described, and the contractors' representative expressing a willingness, I continued the test.

Insulation resistances.—The insulation resistances of the dynamos, cables and switch boards have been found to exceed the requirements.

The "drop" between the machines and the distributing switch board, when the large machines were running at full load, amounted to 12 volts. By throwing the switches out and in, this pressure increased 2 volts for a moment only; when the engine recovered its speed, the voltage dropped to the normal. This is very satisfactory.

The contract has been fulfilled in every particular.

GENERAL REMARKS.

The output of these plants, compared with an equivalent gas light, as obtained from the ordinary bat wing burner, at Washington prices, is favorable. A 5-foot burner gives a light equal to 16 candle power. At one dollar per 1,000 feet of gas, we obtain $16 \times 1{,}000 \div 5 = 3{,}200$ candle power per hour for a dollar.

Taking coal at $2.65 per ton, lamps at 20 cents, repairs at 2

per cent., and allowing an expenditure of 3,000 lamps per year, and a full load on the small plant, the cost per year will be about as follows:

Three engineers,	$3,000
Three firemen,	2,160
One electrician,	1,500
Two linemen,	1,200
Two laborers,	1,000
Repairs,	1,000
Lamps,	600
Coal,	2,453
Oil,	200
Total,	13,113

or $1.50 per hour. The output of the small plant, at the poles, was 57,200 watts per hour. The drop would reduce this about 6 per cent., so that the output at the distributing board would be about 53,768 watts. Taking lamps at 3.2 watts per candle power, the current would give 16,800 candle power for $1.50. Reduced to a dollar, it would be 11,200 candle power for the electric plant, and, as shown above, 3,200 candle power for the gas. The gas output may be doubled per unit of gas by using the Welsbach burner, but the cost of replacing the mantles is considerable, probably half the price of the gas saved.

The output of this electric plant is 228 watts per pound of coal per hour, which, I believe, is a more economical result than is attained anywhere in the District of Columbia. The electric plants in the Capitol are giving excellent results, but they are using Bulkley condensers, the water for which is pumped up by other machinery, the expense for the latter not showing at the Capitol.

One of the local electric lighting companies is charging 13.5 cents per 1,000 watts to private consumers; but to large consumers, particularly those who use most current in day time, it makes a reduction, going as low as 10.7 cents per 1,000. The plant at the Hospital obtains it at 2.8 cents, neglecting interest

on the money invested in the plant, and at 3.5 cents per 1,000, if 6 per cent. interest on the plant is allowed.

Wherever electric is substituted for gas lighting, a greater quantity of light will be used. For example, in the State, War and Navy Department building, the mean number of gas jets in use was 112, before installing the electric plant; since then the mean number of electric lamps used is about 372. Notwithstanding this great increase in the use of light in the building, the aggregate cost so far has not increased.

The consumption of lamps forms an important item in the expense of operating a plant. When I installed the plant in the *Albatross* in 1882, lamps cost $1.00 a piece; now they are bought at 17 cents. With $1.00 lamps, their life time was an important factor; it was economical to obtain great duration at the expense of current. Lamps which glow up to candle power at 3.1 watts per candle power will not last more than half so long as those which consume 4 watts. If we assume 4.3 cents per 1,000 watts for current, and allow 500 hours for each of two lamps, (making 1,000 hours,) the total cost of the lamps, (at 17 cents,) and the current for the 3.1-watt lamps will be $2.47; assuming the same cost for current used in a 4-watt lamp, and taking a single lamp at 1,000 hours, the cost will be $3.09. These figures are from the results I am now obtaining from the plant in the State, War and Navy Department building. It will be seen that the difference in favor of the 3.1-watt lamp is 17 per cent. The total number of lamps in this installation is 2,025, and in the Executive Mansion (which is lighted by the same dynamos), 950, making 2,975 in all. The expenditure of lamps in this building, for the year, was 1,278, or about 63 per cent. of the installation, but during the first part of the year 4-watt lamps were used. The present expenditure is at the rate of 1,800 lamps, or nearly 90 per cent. of the number installed. The central stations figure on two complete renewals a year; they purpose renewing lamps when they glow quite red. My method, in the above building, is to shift the lamps from time to time, from the remote parts of the building to places nearer the dynamos, thus subjecting them to a higher voltage, as there is a drop of 2.5 volts between the

switch board and the farthest lamp. In this way we manage to get a greater life out of the lamps, and yet keep them nearly to candle power.

The mean load factor on one plant in the State, War and Navy Department building, during the year, was about 22 per cent. During office hours it rose to a mean of 36 per cent.; the maximum during first quarter was 53 per cent., second quarter, 66 per cent., third quarter, 114 per cent., and fourth quarter, 73 per cent. A second plant is used to prevent an overload on the first. Our load on Sundays often falls to 12.5 per cent. In electric lighting, as in other commodities, the higher the load factor employed, the greater the economic efficiency.

General Plan

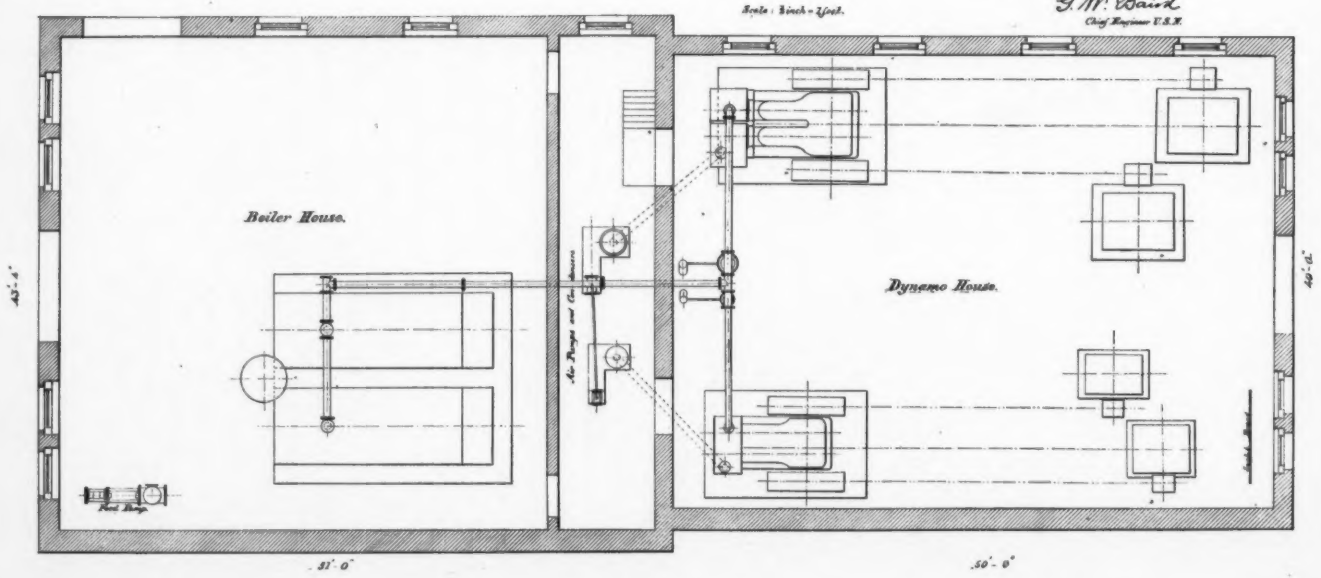
Electric Light Plant.

Government Hospital for the Insane.

District of Columbia.

G. W. Baird
Chief Engineer U.S.N.

Scale: 1 inch = 1 foot.



CONTRACT TRIAL OF THE *GRESHAM.*

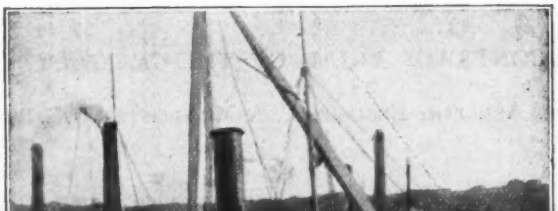
By First Assistant Engineer C. A McAllister, U. S. R. C. S.

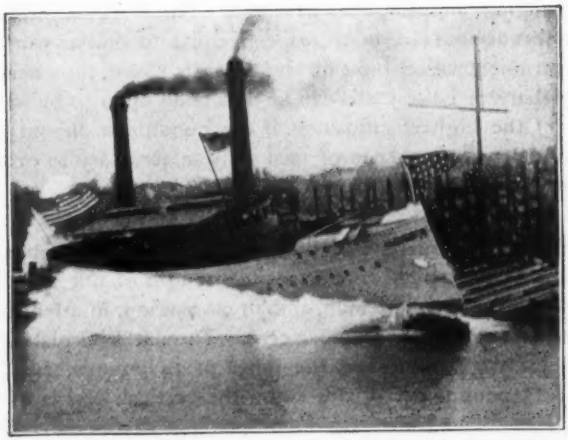
The *Gresham*, a single screw, steel revenue cutter of about 900 tons displacement, was built by The Globe Iron Works Co. of Cleveland, Ohio, from designs furnished by the Treasury Department. The limit of the appropriation was $175,000, and the contract was let at $147,800; as a matter of fact, the actual cost of the vessel was far in excess of that sum. She is intended for service on the Great Lakes, and her headquarters will be at Milwaukee, Wis.

Her rig is that of a two-masted schooner with pole masts, and she is provided with a small sail area. The stem is slightly ram-shaped and provided with a port for a torpedo tube; the stern is overhanging, and elliptical in shape. The workmanship and fittings throughout are in all respects equal to similar work on any government vessel built on the Atlantic Coast, thus demonstrating that our Lake yards can be depended upon to build war vessels of the highest efficiency, if the conditions should ever arise to make the presence of men of war necessary to protect our Northern boundary.

HULL.

The hull is constructed of mild open hearth steel, the physical requirements for which were a tensile strength of not less than 55,000 pounds per square inch, and an elongation, in a length of 8 inches, of not less than 25 per cent. The flat keel plates are worked in two thicknesses, the inner of 14 pounds, and the outer of 15 pounds per square foot; the vertical keel is of 15-pound plate. The transverse frames, except at the extremities of the vessel, are formed of Z-bars, $5 \times 3 \times 3$ inches and 12 pounds per foot, spaced 24 inches between centers. The stem is in three pieces, the two upper ones being of cast steel and the





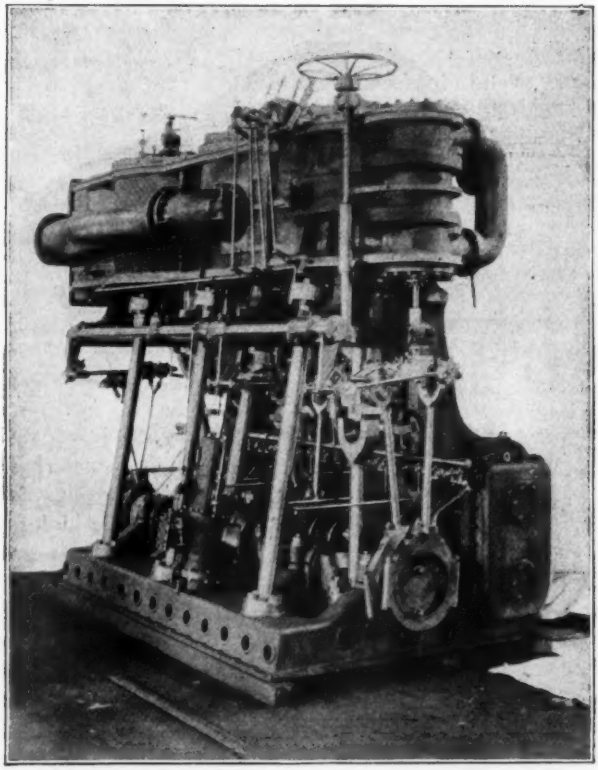
lower of the best hammered wrought iron. The stern frame and rudder frame are of cast steel, each in one casting. The main deck beams are formed of angle bulbs, 6×3 inches and $12\frac{1}{8}$ pounds per foot, with a camber of 7 inches in a length of 32 feet. The outside plating is of 14-pound plate, the sheer strake for a length of 180 feet being double. The hull is divided into eleven water tight compartments. The quarters for officers and crew are roomy, well lighted and ventilated.

Length between perpendiculars, feet	188
over all, feet and inches	205–6
Beam, extreme, at L.W.L., feet	32
Draught, mean, normal, feet and inches	10–9¾
Displacement at this draught, tons	906
Area of immersed midship section, square feet	278.4
Tons per inch of draught	6.91
Coefficient of displacement, prismoidal	.606
midship section	.809
Wetted surface, square feet	6,635

MAIN ENGINE.

There is one vertical, inverted, direct-acting, triple expansion engine, with cylinders 25, $37\frac{1}{2}$ and $56\frac{1}{4}$ inches in diameter, and a common stroke of 30 inches. All three cylinders are fitted with liners, the I.P. and L.P. being steam jacketed. The high pressure cylinder has one piston valve, while the intermediate and low pressure cylinders are each fitted with a double-ported slide valve; all valves are worked by the Stephenson link motion with double bar links. The intermediate and low pressure valves are fitted with balancing cylinders, which connect to the condenser. The cut off for each valve is adjustable by means of slots in the reversing arms.

The reversing gear is of the ordinary direct-acting type, consisting of a steam cylinder bolted to the bed plate, and connecting to double arms on the reverse shaft. The diameter of the reversing cylinder is 12 inches, and its stroke, $12\frac{1}{2}$ inches. In addition to the steam gear there is a screw hand gear, worked by a large hand wheel secured to the high pressure cylinder, convenient to the working platform.



The main pistons are of cast steel, dished, each fitted with two hard cast iron packing rings, $\frac{5}{8}$ inch wide by $\frac{3}{4}$ inch thick. The high and intermediate pressure packing rings are each made in six sections, and the low pressure rings in eight sections. The joints between all sections are made with brass tongue pieces.

The crank and wrist pins are fitted with bolt and nut brasses, which, with the composition eccentric straps, are all lined with Parson's white metal. The crossheads are of forged steel, and bolt to forged steel slippers which are lined on the go-ahead sides with white metal. Each cylinder is supported at the front by one forged steel column 6 inches in diameter. At the back, they are supported on short cast iron housings which are cast with the three sections of the main condenser. Hollow guides for water circulation are bolted to these back columns. The bed plate is of cast iron, in one section, which bolts to the condenser and engine foundation. The top of the engine foundation is formed of one steel plate, $\frac{3}{4}$ inch thick, which continues aft and forms the top of the foundation for the thrust bearing. All steel forgings used about the engine are of the best quality obtainable, and were subjected to a most rigid inspection. The physical requirements for the piston and connecting rods, valve stems, crossheads, etc., were a tensile strength of not less than 80,000 pounds per square inch, and an elongation, in 2 inches, of not less than 22 per cent. For the front columns and shafting, a tensile strength of not less than 60,000 pounds per square inch, and an elongation of 25 per cent. in a length of 8 inches were required.

The lubricating system is thorough and complete, nearly all moving parts being oiled automatically. Oil for the crank pins, wrist pins, main bearings and eccentrics, is taken from three overhead reservoirs, secured to the sides of cylinders overhead.

The following are some of the principal dimensions, ratios, etc., of the engine.

Number of cylinders	3
Diameter of H.P. cylinder, inches	25
I.P. cylinder, inches	$37\frac{1}{2}$
L.P. cylinder, inches	$56\frac{1}{4}$

Stroke of pistons, inches		30
Valves, diameter of H.P., inches		16
length of I.P., feet and inches		2-11 11/16
width of I.P., feet and inches		3-0½
length of L.P., feet and inches		3-4⅝
breadth of L.P., feet and inches		4-8
diameter of L.P. balance piston, inches		5½
effective area of I.P. balance piston, inches		21.35
diameter of L.P. balance piston, inches		9
effective area of L.P. balance piston, inches		61.21
Area H.P. steam ports, maximum opening, inches	top	59.96
	bottom	64.12
H.P. exhaust ports, maximum opening, inches	top	76.00
	bottom	76.00
I.P. steam ports, maximum opening, inches	top	101.00
	bottom	108.00
I.P. exhaust ports, maximum opening, inches	top	128.00
	bottom	128.00
L.P. steam ports, maximum opening, inches	top	202.00
	bottom	220 93
L.P. exhaust ports, maximum opening, inches	top	293.87
	bottom	293.87
Valve stems, H.P., diameter, inches		2
H.P., diameter through valve, inches		1½
I.P. diameter, inches		2½
I.P., diameter through valve, inches		2⅛
L.P. diameter, inches		2½
L.P., diameter through valve, inches		2⅛
Main steam pipe (9 inches in diameter), area cross section, square inches		63.61
Exhaust pipe from I.P. to L.P. (14½ inches in diameter), area cross section, square inches		165.13
Exhaust pipe to condenser (19 inches in diameter), area cross section, square inches		283.53
Volume swept by H.P. piston per stroke, cubic feet		8.522
I.P. piston per stroke, cubic feet		19.175
L.P. piston per stroke, cubic feet		43.142
Ratio of net area of H.P. to I.P. piston		1:2.26
I.P. to L.P. piston		1:2.25
H.P. to L.P. piston		1:5.06
Piston rods, diameter, inches		5
length from piston to crosshead, feet and inches		4-1¼
Connecting rods, length, center to center, feet and inches		6-3
diameter of upper end, inches		5
lower end, inches		6½
Crossheads, surface (go-ahead), square inches		208
Crosshead pins, diameter, inches		5¾

Crosshead pins, length, inches	6½
Reversing gear, diameter of cylinder, inches	12
stroke of piston, inches	12½
Crank shaft, diameter, inches	10½
coupling discs, diameter, inches	21
thickness, inches	3
journals, diameter, inches	10½
length, inches	13½
Crank pins, diameter, inches	10½
length, inches	13½
Crank webs, width, inches	12
thickness, inches	6½

SHAFTING.

Diameter of thrust shaft, inches	9¾
Number of thrust collars	9
Diameter of thrust collars, inches	15½
Thickness of thrust collars, inches	1¾
Space between collars, inches	3½
Actual thrust surface, square inches	575.04
Diameter of couplings, inches	29
Thickness of couplings, inches	3
Number of bolts in each flange	6
Diameter of bolts, inches	2⅜
Length of thrust bearing, feet and inches	5–5½
Diameter of intermediate shaft, inches	9¾
Length of intermediate shaft, feet	28
Diameter of propeller shaft, inches	10½
Length of propeller shaft, feet and inches	26–10
Total length of composition casing, feet and inches	22–8¾
Thickness of composition shaft casing at bearings, inch	¾
elsewhere, inch	½
Length of stern bearing, feet	4
inboard bearing, feet	3

PROPELLER.

There is one solid, right-handed, four-bladed, true screw propeller, of manganese bronze. The ends of the blades at the periphery are inclined 12 inches aft of the center of the hub. It is secured in place by a feather key, $2\frac{1}{4} \times 1\frac{3}{8}$ inches, and a steel nut screwed on, and secured from backing off by a key driven through the end of the shaft. The physical requirements for the manganese bronze were an average tensile strength of 55,000 pounds per square inch, an elastic limit of at least 18,000 pounds, and an elongation of not less than 18 per cent. in 2 inches.

Number of blades	4
Diameter of propeller, feet	10
Pitch, true, feet and inches	12– 6
Greatest width of blade (3 feet from center of hub), feet and inches	2–11 $\frac{3}{16}$
Length of hub, inches	23
Helicoidal area, square feet	40.08
Disc area, square feet	78.54
Ratio, pitch to diameter	1.25
disc area to area of immersed midship section	.28
Immersion of center of propeller at draught of 11 feet 6 inches, feet	6

MAIN CONDENSER.

The condenser is of cast iron in three sections, which form a part of the engine framing. The circulating water passes through the tubes and the exhaust steam around them. Connections are provided for the main and auxiliary exhaust pipes, evaporator, supplementary feed, soda cock and boiling out nozzle. The condenser will also be used as an auxiliary condenser, and for this purpose, a suction pipe is connected from the auxiliary feed pump to the bottom of the air pump channelway.

Length over all, feet and inches	16–5 $\frac{1}{4}$
Width of shell, inside, feet and inches	2– $\frac{1}{2}$
Thickness of shell, inch	1
Length of tubes between tube sheets, feet and inches	14–5
Number of tubes	1,253
Outside diameter of tubes, inch	$\frac{5}{8}$
Thickness of tubes	No. 18 B.W.G.
Condensing surface, square feet	2,962.8
per horse power, square feet	1.22

PUMPING OUTFIT.

There is one double, vertical, single-acting air pump of the Blake type. The two pumps are connected by beams pivoted at the center, which receive their motion from the crossheads on the pump and piston rods by means of links. The pump is bolted directly to the bottom of the condenser. The steam cylinders are 9 inches in diameter, the air cylinders, 20 inches in diameter, and the stroke, 12 inches.

The circulating pump is of the centrifugal type, operated by a vertical engine, 8 inches in diameter and 8 inches stroke. The pump suction is 12 inches, and the delivery, 10 inches in diam-

eter. There are two main injection valves, one at the bottom and the other at the side. There is also a bilge injection valve, fitted with an automatic closing device.

The main and auxiliary feed pumps are of the Blake navy vertical, duplex type, with steam cylinders 8 inches in diameter, water cylinders 5 inches in diameter, and a stroke of 12 inches.

The fire pump is of the same type as the feed pumps, with 14-inch steam cylinders, 8½-inch water cylinders and a stroke of 12 inches.

There are also in addition to the above a distiller circulating pump, combined fresh and salt water pump, a bilge pump, and an air pressure pump for the water tanks, all of the Blake pattern.

DISTILLING APPARATUS.

Although this vessel is intended for service on the fresh water of the Great Lakes, a distilling apparatus capable of producing 3,500 gallons of potable water in twenty-four hours is supplied. The object of fitting the distilling apparatus is so that it can be used in case the vessel is transferred for service on the Atlantic or Pacific coasts. With this possibility in view, the same precautions have been taken in fitting zinc rings around the openings for sea valves, etc., as would be taken for a vessel built for service on salt water.

ELECTRIC LIGHT.

The vessel is lighted throughout with incandescent electric lights, and in addition, there is a search light located on the bridge. The plant consists of one General Electric Co.'s dynamo, of 10 kilowatts capacity, at 80 volts and 125 amperes, which is operated by a direct-connected, vertical, double engine, designed for a working pressure of 80 pounds. The cylinders are 6½ inches in diameter and 5 inches stroke. There are 139 16-c. p. fixed incandescent lights, located in the various parts of the vessel. The search light is of 25 ampere capacity.

OTHER AUXILIARIES.

There are in addition to the above, a steam steering engine, steam windlass and a See hydro pneumatic ash chute. There is

a small workshop containing a lathe, drill and shaper, all run by a vertical engine.

BOILERS.

There are four single-ended boilers of the Scotch type, each having two corrugated steel furnaces. The four boilers are arranged in pairs in one water tight compartment, with an athwartship fireroom between them. One of the leading features of these boilers is the shell of each, which is made of two plates only, thus doing away with the central circumferential seam. The ordered dimensions of these shell plates were 217½ inches × 115⅛ inches × 1 1/16 inches, the calculated weight of which is 7,642 pounds. The slabs from which these plates were rolled averaged each 10,000 pounds in weight.

The longitudinal seams are treble-riveted, and fitted with outside and inside butt straps. The circumferential seams are lapped and double-riveted.

The forced draft is on the closed fire room system, the air being furnished by two Sturtevant blowers. The fans are driven by enclosed, two cylinder, vertical, simple engines, with cranks set opposite each other. Only one piston valve is used for both cylinders. Air locks are provided in the passage way leading to the fire room, and in the ventilators, for hoisting ashes when under forced draft.

Length over all, feet	10
Diameter, outside, feet and inches	11–6
Number of furnaces	2
Diameter of furnaces, outside, feet and inches	3–10
inside, feet and inches	3–6
Length of grate, feet	6
Outside diameter of tubes, inches	2½
Length of tubes between sheets, feet and inches	7–3
Spacing of tubes horizontally, inches	3¾
vertically, inches	3½
Number of ordinary tubes	178
stay tubes	46
Thickness of ordinary tubes, B.W.G.	10
stay tubes, B.W.G.	6
Total number of tubes in one boiler	224
Number of combustion chambers	2

Depth of combustion chambers, inches	23½ and 24½
Width of combustion chambers, feet and inches	4–10
Thickness of shell plates, inches	1 1/16
heads, top and bottom, inch	3/4
tube sheets, front and back, inch	3/4
combustion chamber plates, inch	½ and 5/8
furnaces, inch	9/16
Diameter of rivets in longitudinal seams, inches	1⅛
circumferential seams, inches	1⅛
Pitch of screw stays, inches	7 × 7
Diameter of screw stays, inches	1 5/16
Number and diameter of through braces above tubes, inches	16 of 2¼
braces to back tube sheets, inches	2 of 2
around lower manhole, inches	1 of 2 and 2 of 1¾
Tube heating surface, one boiler, square feet	1,080.5
Plate heating surface, square feet	215.8
Total heating surface, square feet	1,296.3
Grate surface, square feet	42
Area through tubes, square feet	5.94
over bridge walls (two), square feet	6.4
through uptake, square feet	8.23
Volume of furnaces and combustion chambers above grates, cubic feet	160.8
Steam room, water six inches above tubes, cubic feet	235.52
Water surface, water six inches above tubes, square feet	99.94
Diameter of smoke pipe (inside), feet and inches	6–3
Area of smoke pipe, square feet	30 68
Height of smoke pipe above grates, feet	54
Diameter and number of safety valves, one boiler, inches	2 of 3
Load on safety valves, pounds	160
Diameter of main stop valves, inches	5
auxiliary stop valves, inches	4
Totals for four main boilers:	
Tube heating surface, square feet	4,322
Plate heating surface, square feet	863.2
Total heating surface, square feet	5,185.2
Grate surface, square feet	168
Area through tubes, square feet	23.76
Area over bridge walls, square feet	25.6
Area through uptakes, square feet	32.92
Area of water surface, square feet	399.76
Volume of steam room, cubic feet	942.08
furnaces and combustion chambers above grates, cubic feet,	643.2
Ratios:	
Tube H.S. to G.S.	25.72 to 1
Plate H.S. to G.S.	5.14 to 1
Total H.S. to G.S.	30.86 to 1

Ratios:	
Area through tubes to G.S.	.14 to 1
of water surface to G.S.	2.38 to 1
Steam room per square foot of G.S., cubic feet	5.60
Volume of combustion chambers and furnaces above grates and bridge walls per square foot of G.S., cubic feet	3.83

TRIAL TRIP.

The official contract trial took place on December 11, 1896. No requirements for speed were mentioned in the specifications, although it was expected that the vessel would make 16 knots, with the main engine developing 2,000 horse power. The machinery specifications stipulated that the main engine should be run under maximum conditions, with the vessel underway, for a period of eight hours. The trial of the steam machinery was conducted by a board consisting of Engineer-in-Chief J. W. Collins, Chief Engineer J. E. Jefferis, First Assistant Engineers C. A. McAllister and C. M. Green, and Second Assistant Engineer J. Q. Walton, U. S. R. C. S.

The speed was recorded by a patent taffrail log, the accuracy of which was tested by runs over known distances, developing the fact that the apparent speed was slightly below the actual speed obtained. The course run was in a northeasterly direction from the entrance to Cleveland harbor, which gave a head wind for the first part of the trial. Owing to a considerable variation in the depth of water, which at the best was too shoal for a trial trip, the speed varied at different parts of the course, although the engine was developing practically the same power.

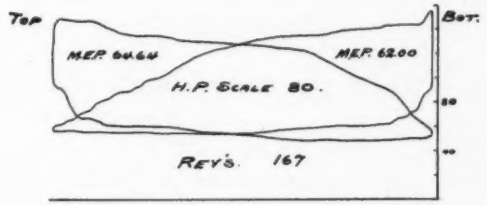
The trial was a marked success in every way. The boilers steamed freely, and the main and auxiliary engines worked without a heated bearing. The vibrations of the hull at full speed were scarcely perceptible. The maximum speed obtained, 17.53 knots (mean for one-half hour), in comparatively shoal water, places the *Gresham* in the front rank of vessels on the Great Lakes, so fas as speed is concerned. With the possible exceptions of the *North West* and *North Land*, also built by the Globe Iron Works Co., it is doubtful if there is another vessel on the lakes capable of equaling the *Gresham's* speed.

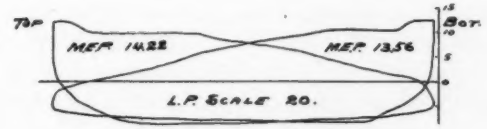
DATA OF TRIAL.

Draught, mean, for trial... { forward, ft. and ins.... 9–2 / aft, ft. and ins..... 10–8 } ft. and ins.....	9–11
Displacement at mean draught on trial, tons	820
Area of immersed midship section, square feet	251.8
Wetted surface, square feet	6,291
I.H.P. (total) per 100 square feet of wetted surface	26.8
Average speed, in knots	17.32
Slip, per cent	15.05
Speed^3 × area immersed midship section ÷ I.H.P.*	557
Speed^3 × displacement$^{\frac{2}{3}}$ ÷ I.H.P.*	191.8
Revolutions of main engine per minute	165.3
Piston speed, in feet per minute	826.5
Steam pressure at boilers, per gauge, pounds per square inch	159
engine, per gauge, pounds per square inch	152
in first receiver, pounds per square inch	52
second receiver, pounds per square inch	10
Vacuum in condenser, inches of mercury	25
Opening of throttle	Wide.
Steam cut-off in fraction of stroke from beginning, H.P.	.617
I.P.	.701
L.P.	.680
Double strokes of air pump per minute	44.2
feed pump per minute	48.3
Revolutions of circulating pump per minute	186.5
Temperatures in degrees Fahrenheit, engine room	81.3
injection	36.0
discharge	112.6
feed	108.6
fire room	78.2
Average air pressure in fire room, inch of water	.84
Revolutions of blowers per minute	611
Mean pressures in cylinders, H.P.	55.48
I.P.	31.02
L.P.	13.49
aggregate equivalent, reduced to L.P.	37.8
I.H.P., H.P. cylinder	666.5
I.P. cylinder	848
L.P. cylinder	832.5
total for main engine	2,347
air pump, estimated	10.6
circulating pump, estimated	8.3
feed pump, estimated	18.2
blowers, estimated	42.4

* For main engine only.

I.H P., other auxiliaries, estimated	4.5
total for all auxiliaries	84
machinery in operation	2,431
Indicated thrust, main engine, pounds	37,552
per square foot of developed area (40.08 square feet) of propeller, pounds	937
per square inch of thrust surface, pounds	65.3
Cubic feet swept per minute by L.P. piston per I.H.P	6 05
Square feet of cooling surface per I.H.P	1.218
heating surface per I.H.P	2.13
I.H.P. per square foot of G.S. (all machinery in operation)	14.47
Maximum performance (one-half hour):	
Speed in knots	17.53
Average revolutions, main engine	167.5
I.H.P. for main engine (average)	2,497.06
auxiliaries, estimated	91
Total I.H.P., main engine and auxiliaries	2,588.06
Air pressure in fire room, inch of water	92





Total I.H.P. 2515

Cards No. 34 - 2.00 P.M.

Steam Pressure Engine Gauge	150
Revolutions	167
1st Receiver	58
2nd Receiver	11
Vacuum	24

U.S.S. "GRESHAM"

Cylinders - $35'' \cdot \frac{87\frac{1}{2}}{30} \cdot 54\frac{1}{2}''$

Lake Erie ---------- Dec. 11th 1896

EXPERIMENTS WITH AN AIR-LUBRICATED JOURNAL.

By Professor Albert Kingsbury, Associate Member.

In a paper read before the American Society of Mechanical Engineers, December, 1895,* the present writer mentioned a case in which air was observed to act as an effective lubricant.

A trunk piston, 6 inches diameter, 6 inches long, was carefully fitted to its cylinder. The diameter of the cylinder was about .0008 inch greater than that of the piston. When the axis was horizontal, the piston could be revolved by hand at a rate sufficient to make the air in the thin space between piston and cylinder act as a perfect lubricant, forming a complete separating film between the metallic surfaces, and reducing the friction to an extremely small amount. The weight of the piston was 22 pounds, the casting being hollow.

The phenomenon thus observed was afterwards turned to advantage in the construction of an apparatus for exhibiting the action of fluid lubricants in cylindrical bearings. This apparatus (to be presently described) was exhibited before the Mechanic Arts Section of the Association of Agricultural Colleges and Experiment Stations, at Washington, D. C., November, 1896. At that time no extended series of measurements had been made with the apparatus, and the results that had been obtained were qualitative rather than quantitative. Since that time a large number of tests have been made with the apparatus at various speeds, measuring the pressures in the lubricant, thickness of the film of lubricant, and the friction. A statement of these results is included in the present paper.

The apparatus serves to demonstrate chiefly:

* Transactions Am. Soc. M. E., Vol. XVII, p. 116.

1. The formation of a film of the lubricant (air) completely separating the revolving plug from contact with the ring.

2. The position which the plug takes in the bore as due to the formation of the film, and the way in which this position changes with the speed of rotation.

3. The distribution of pressures in the film of the lubricant at various rates of rotation.

The employment of atmospheric air as a lubricant is by no means a novelty. It was mentioned by Hirn* as long ago as 1855. He says: "Air under certain circumstances and in sufficient quatities becomes the best lubricator, the coefficient then falling to $\frac{1}{10000}$." The lubricating effect of air between two well-fitted surface plates is frequently observed. But cylindrical journals in which air lubrication is possible are by no means common.

It will of course be understood that it is no part of the object of this paper to advocate the employment of air for ordinary purposes of lubrication, although there might be cases in which it could be thus used advantageously. It was employed in the experimental apparatus only because of its peculiar adaptability to the purposes of exhibition. The atmosphere furnishes a bath of the lubricant, thus maintaining a constant supply of constant quality. It reduces the friction so greatly that the apparatus once set in motion by hand will continue to run for several minutes, thus making the apparatus serviceable without a motor. Air being a non-conductor, it is possible to use the electric current for proving the existence or non-existence of the separating film.

To the novice it is sufficiently startling to learn that even the comparatively heavy and viscous oils may completely separate the rubbing surfaces of a bearing. A convincing demonstration that a fluid of so low viscosity as air may be made to form a separating film in a bearing carrying an appreciable load, aids one in understanding the effects of the more viscous lubricants. Indeed, it is not necessary to use so "heavy" a lubricant as air for demonstration. The apparatus to be described has been run

* "Engineering," January 30, 1885, p. 118.

quite successfully with hydrogen as a lubricant, the dry gas being delivered from a generator to an "oil hole" in the top of the bearing. In this case the friction was apparently much less than that due to air, though a higher speed was necessary for the maintenance of the film of hydrogen.

DESCRIPTION OF THE APPARATUS.

The apparatus, as mounted for the purposes of demonstration, is shown in Fig. 1, also in the drawings (Figs. 2*a* and 2*b*). The essential parts are the cast iron ring or sleeve *R* and the steel plug or short piece of shaft *P*. This plug is 6 inches in diameter, 6¼ inches long, and its weight, 50½ pounds, is the "load" of the

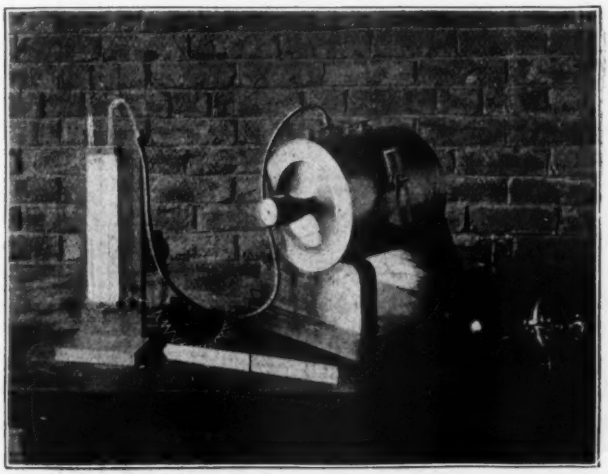


FIG. 1.

bearing. The bore of the ring and the cylindrical surface of the plug are carefully finished by grinding processes, so that they are very nearly true cylinders; but not as nearly true as could be attained by hand fitting, if the bore of the ring could be thus treated. The diameters of the two surfaces differ on the average by about sixteen ten-thousandths of an inch. The sleeve

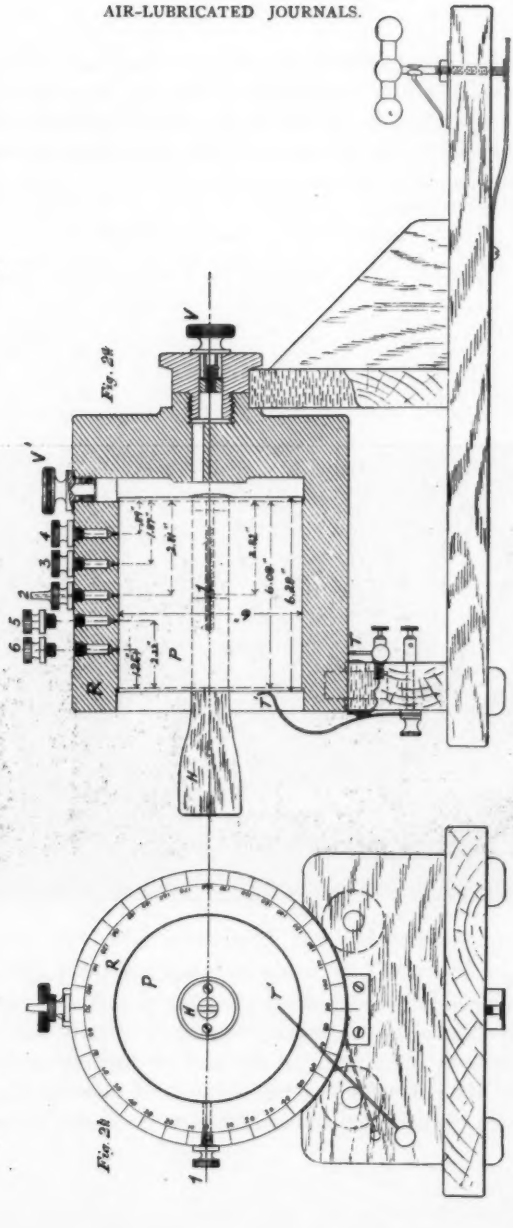


Fig. 2h

Fig. 2h

is closed at one end. Two valves V, V', are provided, one of which is opened to permit the escape of the air while the plug is being inserted. When in place the plug is about $\frac{1}{4}$ inch from the head; then, the valves being closed, the end motion of the plug is limited by the confined air. The plug is set into rotation by means of the wooden handle H; higher speeds are obtained by an ordinary geared hand drill (shown in Fig. 1), adapted to connect with the slot in the iron plate in the end of the handle H.

The ring is mounted on a wooden holder, as shown, so that it may be turned about its axis, bringing any one of the test holes (numbered 1 to 6) to any desired position angle, the settings being made by the graduations on the face of the ring.

The auxiliary apparatus consists of a mercurial manometer, which may be connected with any one of the test holes by a rubber tube and a nipple; also a galvanic cell with a signal bell in circuit, one terminal being in contact with the ring at T; the other terminal may be brought into contact with the plug at T'. The manometer, bell and cell are mounted on a stand, as shown in Fig. 1.

METHODS OF DEMONSTRATION.

1. *The formation of the film of the lubricant.*—When the plug is not in motion, its weight is undoubtedly supported by metallic contact with the ring at the bottom of the bearing, both surfaces being perfectly clean and dry. This is evidenced (if necessary) by a familiar grating sound if the plug is turned slowly; also by the ringing of the bell if the wire T' be pressed against the plug. The usual friction due to such contact may also be shown by the sudden stopping of the plug after being set into slow motion. But when the motion is more rapid (not less than about 200 revolutions per minute) an entire change of condition is evident. The sound ceases; there is apparently very little friction, as the plug continues to revolve by its own inertia for several minutes. If now the wire T' be pressed against the plug, no current passes. This last is most conclusive proof of the existence of the film of air, maintained, as in the case of other fluid lubricants, by virtue of the viscosity of the fluid and its adhesion to the surfaces of the bearing, which are in relative motion.

2. *The position of the rotating plug in the bore of the ring.*—It has been stated that the difference of diameter of plug and bore is about $\frac{16}{10000}$ inch. This small difference is essential to the formation of the film and the consequent separation of the surfaces. But the film when formed is not of the same thickness at all parts; the plug is not in the center of the hole. To show this, a small screw of fine thread is inserted through the hole marked No. 1, as indicated at *S*, Fig. 3. While the plug is revolving, the screw is turned until its point barely touches the plug, as shown at *a*. The contact is indicated by the signal bell. If now, without farther adjustment of the screw, the ring is turned

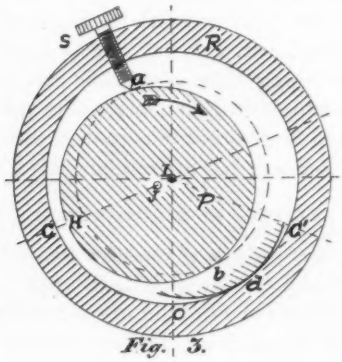


Fig. 3.

slowly toward the right, the contact is lost, but is formed again when the screw comes into the position as shown at *b*. If the ring is turned in the opposite direction from *a*, the pressure of the contact increases, as shown by increased energy of the current at the bell. Evidently, then, the plug is not central in the bore, and the surfaces are nearest at a point *G* found by bisecting the arc *ab*. If the screw is placed in contact at *G*, the contact is lost if the ring is turned in either direction from that position. If the plug rotates to the right, *G* is on the left of the vertical diameter; if the rotation is to the left, *G* assumes a position *G'* on the right of the vertical. The angle *GJO* varies with the speed, but this is not easily shown without some means

for maintaining constant speed at any desired rate. The results of measurements of the variation of the angle GJO are shown in Fig. 14.

The axes of plug and ring approach each other as the speed of rotation of the plug increases. To show this, the screw is set barely in contact at G, the plug turning to the right. If now the speed of the plug is increased, the contact is lost; as the speed again decreases by reason of friction in the air film, the contact with the point of the screw is formed again. The curve (Fig. 15) shows the results of measurements of this variation.

The position which the plug assumes in the bore of the ring, when supported by the film of air which is maintained by the rotation, is in accord with the results of theory and experience. It is stated by Mr. Wm. Kent* that the brasses of car journals always wear most on the "forward" side. If the action of the lubricant is not considered, this wear takes place where it might least be expected. It is ordinarily assumed that when rotation is just beginning, the frictional resistance causes the shaft to roll up the side of the bearing a short distance before sliding begins; this would place the contact at d (Fig. 3), on the opposite side of O, from where G is found in these experiments. It is probable that the rolling and sliding do take place in any case when the motion is just beginning, but immediately the pressure set up in the lubricant as the shaft draws it into the wedge-shaped space above d forces the shaft into the position of equilibrium in which it has been found in the experiments. If the film be complete, it is reasonable to suppose that the only wear will be due to solid particles present in the lubricant, and this wear would be greatest on the ring near the point G. Again, if too great a load be on the bearing, or conversely, the speed of rotation or the viscosity of the lubricant be too low, the film will be incomplete; the metals may come into contact, but such effect as the lubricant still has will be felt on the right of O (Fig. 3), thus concentrating the wear on the left. This latter condition must exist, for a short time at least, whenever the motion is starting or stopping; and it is probable that in many cases this is the chief cause of wear. If

* Mechanical Engineers' Pocket-Book, p. 937.

the viscosity of the lubricant be barely sufficient to maintain the complete film at the normal speed of working, then a greater amount of wear will take place than if a more viscous oil were used; although the average power required to drive the journal (overcoming fluid friction only) may be greater in the latter case. "We found that the oils that gave us the best results in the economy of power were the ones that produced the most rapid wear."* (On mill spindles.)

The relative positions of the cylindrical surfaces, as shown in the experiments, vary in accordance with the theory of Professor Osborne Reynolds.† An elastic fluid lubricant was used in the experiments, while the theory is derived for incompressible fluids; but the direction of variation of the axial distance and of the position of the thinnest point of the film are, as far as observed in the experiments, the same as the theory indicates.

3. *The distribution of pressures in the film.*—The existence of the separating film implies that a distribution of pressures exists in it such that the sum of the vertical components (friction being disregarded) equals the weight of the plug. To examine the pressure distribution, a nipple is screwed into one of the test holes (numbered 1 to 6, Fig. 2*a*). From the nipple a small rubber tube leads to the U-shaped mercurial manometer. The indications of the manometer are in pounds per square inch above or below atmospheric pressure.

By turning the ring to bring the test hole into any desired position, the pressure may be noted at any point round the bearing; each new position of the test hole being equivalent to "tapping" the film in a new place. The photograph Fig. 1 was taken with the plug in "clockwise" rotation; the test hole is near the point of minimum pressure, and the manometer shows the mercury higher in the arm of the tube connected with the film.

The distribution of the pressures varies with the speed. In Fig. 6 are plotted the results of the measurements of pressures at the middle of the bearing, the pressure being laid off to scale radially from the circle representing atmospheric pressure. This

* Dr. F. W. Arvine. Trans. Am. Soc. Mech. Eng., Vol. VI, p. 162.

† "Theory of Lubrication"—Phil. Trans. (Royal Society, London.) 1886.

circle may also be taken to represent the middle section of the bearing. The pressure at any point is laid off at that point.

The curves show the pressures for 230 and for 1,730 revolutions per minute. It is seen that although these curves differ considerably, the negative pressures are practically all above the horizontal diameter, the positive pressures, below; and the maximum and minimum pressures are both on the left of the vertical (the rotation being negative or "clockwise"). The approximate symmetry of the pressure distribution is better shown in the curves from the same data (Fig. 9) to rectangular co-ordinates; a third curve (805 R.P.M.) is shown in the same figure.

Since the principal points of these curves (the points of maximum, of minimum and of atmospheric pressure) do not shift very greatly with moderate changes of speed, it is quite possible to show the general distribution of pressure when the apparatus is driven by hand. The curves shown were plotted from data obtained when the plug was run by power at constant speeds. The position of the maximum pressure has a general correspondence with that found in the well known experiment of Mr. Beauchamp Tower,* which experiments were made with a brass encircling not quite one-half the circumference of the shaft, and corresponding in direction of pressure to the lower half of the present apparatus.

The existence of a region of pressure lower than that of the atmosphere has been noted in experiments upon oil lubricated bearings. In a paper by Mr. John Dewrance, Associate Member of the Institute of Civil Engineers (of which an abstract is given in the "American Machinist," August 27, 1896), an instance is given in which a "vacuum of 30 inches" was noted; and in the same paper a maximum pressure of 3,000 pounds per square inch is recorded.

In any actual bearing in which the maximum oil pressure may be several hundreds of pounds per square inch, the negative pressure, if existing, would probably be small by comparison with the positive and of little or no value as aiding to support the load. In the present experiments, the reduction of pressure

* Proceedings Inst. Mech. Engineers. January, 1884.

on the top of the shaft is of nearly equal value with the increase of pressure on the lower side, both being effective for support of the load.

The existence of regions of "positive" and "negative" pressures in the film points at once to some practical considerations of the introduction of lubricants into bearings. If free openings were made near the minimum and the maximum pressure points in the ring, air would evidently flow in at the one and out at the other. Hence, to make the bearing take up its lubricant freely and automatically, it must be fed at a point where the pressure is not greater than that of the atmosphere; and no openings should be made in the region of positive pressure. These conditions are easily fulfilled in a bearing in which the pressure is always in nearly the same direction; changes of pressure direction may complicate the problem. Much valuable information on this matter of the application of oils is contained in two recent papers by Mr. Dewrance, one before the Institute of Civil Engineers, the other before the Institute of Marine Engineers. These papers are republished in the last number of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. In the same number of the JOURNAL (Vol. IX, No. 1) is one of a series of papers "On Crank Shafts," by Mr. John H. Macalpine, in the course of which he gives a concise digest of the papers of Tower, Reynolds and Dewrance above referred to; also some interesting experience of his own in this matter of making the conditions such that the lubricant may be most effective.

It is essential to perfect lubrication that the shaft should be appreciably smaller than the bore, if the latter embrace an arc approaching or exceeding 180 degrees. This was experimentally verified in the process of fitting the plug to the ring. When the plug entered perfectly freely but without any "shake" it was impossible to make it run as described. In the case of a brass embracing less than one-half the circumference of the shaft (as in a car journal) there is an opportunity for the oil to lift the brass from the journal. Thus the necessary wedge-shaped space is formed, even though the brass and the axle be of exactly the same radius. The yielding of the brass under the pressure, and

its expansion by the heat developed by friction, may aid in effecting the same result.

METHODS FOR MORE EXACT INVESTIGATION.

As has been stated, the operation of the apparatus by hand serves most of the purposes of exhibition; but for quantitative results constant speed at any desired rate is necessary. To obtain the results plotted in the diagrams, Figs. 6 to 11, 14 and 15, the apparatus was mounted in a lathe as shown in Fig 4. The plug was driven from the spindle by the slender rod or wire, $\frac{1}{8}$ inch to $\frac{3}{16}$ inch diameter, the flexibility of which prevented the transmission of any sensible external pressure to the plug.

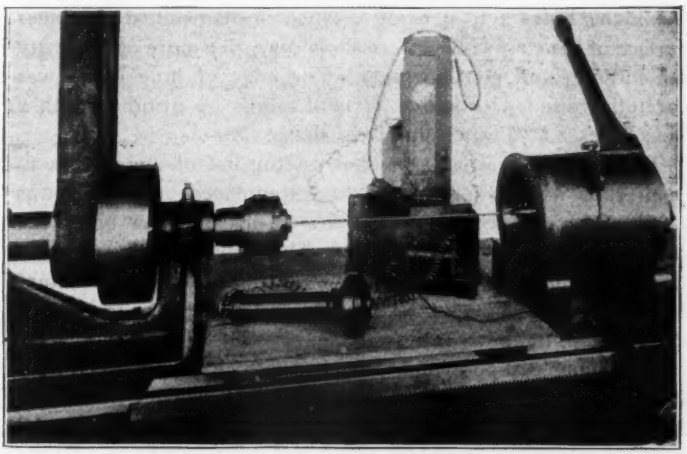


FIG. 4.

The pressure observations were made as previously described, taking readings for points 20 degrees apart around the bearing; first with the plug rotating in one direction, then in the opposite direction, reversal being obtained by crossing the over head belt. Usually the pressures obtained in one direction were higher than the corresponding ones in the other direction, by a nearly constant amount for any given test hole and given rate of rota-

tion. This amount varied in the several cases from a very small quantity up to one pound per square inch. This is thought to be due to the irregularities in the surface. If a test hole is situated on the side of a "hill" in the surface, it might be expected that the results would differ as stated. In all cases the curves are plotted as the mean of the curves taken in opposite directions of rotation; the angular position of the test hole being referred to *O* (Fig. 3), and measured in the direction of rotation. Test hole No. 1 gave curves showing an excess of pressure over those of No. 2, the excess being nearly uniform all around the bearing for any speed, and increasing with the speed, so that at 4,400 R.P.M., the negative pressures nearly disappeared for hole No. 1. This is believed to be due to the larger diameter of hole No. 1 ($\frac{1}{32}$ inch), holes 2 to 6 being $\frac{1}{16}$ inch in diameter at the inner surface of the ring. The larger hole may give more opportunity for eddying and inertia effects. The edge of hole No. 1 was carefully rounded to about $\frac{1}{32}$ inch radius by grinding with a lap; holes 2 to 6 were left nearly sharp.

The usual difficulties attendant on the use of mercury in an open tube as a manometer were experienced. Fouling was finally overcome by placing a few drops of water upon the mercury in each arm of the tube. This protected the mercury from oxidation, and the meniscus remained perfectly constant as the pressure changed. The water did not appear to lose by evaporation, the bore of the tube being only $\frac{1}{10}$ inch.

The changes of position of the plug with change of speed were determined after the general method already given. In place of a simple contact screw as shown in the diagram Fig. 3, a special micrometer (shown in Fig. 2*c*) was used. The screw was $\frac{1}{8}$ inch diameter, 40 threads; the arm was graduated to $\frac{1}{10000}$ inch. This was tested for its total range of measurement (about $1\frac{1}{2}$ thousandths of an inch), and found to be very nearly correct. The signal bell was discarded for a more sensitive means of detecting contacts. A hand telephone was placed in the secondary of a small Ruhmkorff coil, the ring and plug being virtually the terminals of the primary circuit. The rotating plug being more or less rough, and not exactly round, formed its own "make-

and-break" with the screw point at each revolution, giving a very sharp indication in the telephone. A weak current was used. From numerous tests it is believed that the method will indicate contact variations much smaller than .00001 inch, possibly one-half that amount.

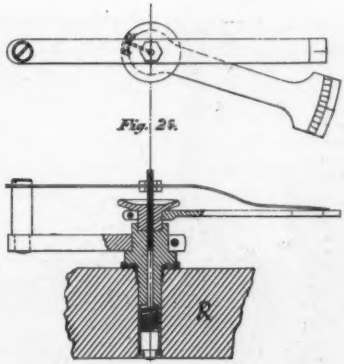


Fig. 25.

The position of the thinnest point of the film was determined as already indicated. Then the screw was set at this point (*G*, Fig. 3), a reading taken; then the screw was set at a point diametrically opposite *G*, a reading again taken. This was repeated three or four times. The difference of mean readings divided by 2 gave the distance between axes of plug and ring. These observations were taken for both directions of rotation. For the position of the thinnest point of the film, the mean of each set of results is shown in Fig. 14, together with the mean of both. For the axial distances, the mean results obtained in opposite directions of rotation never differed more than .00003 inch, and in some cases agreed within less than .00001 inch, the higher speeds giving the closer agreements usually. The mean distances between axes are shown in Fig. 15.

The lathe shown in Fig. 4 could be driven only up to 1,730 R.P.M. For the higher speeds the apparatus was mounted in another lathe, as shown in Fig. 5. The ring was here carried on the lathe spindle, and the plug was driven from the spindle

of the small grinding device mounted on the lathe carriage. The photograph does not show this spindle connected for driving the plug, but shows the plug connected with a wire to the tail stock, for the friction test to be described later. The pressures, etc., at the higher speeds were measured as before.

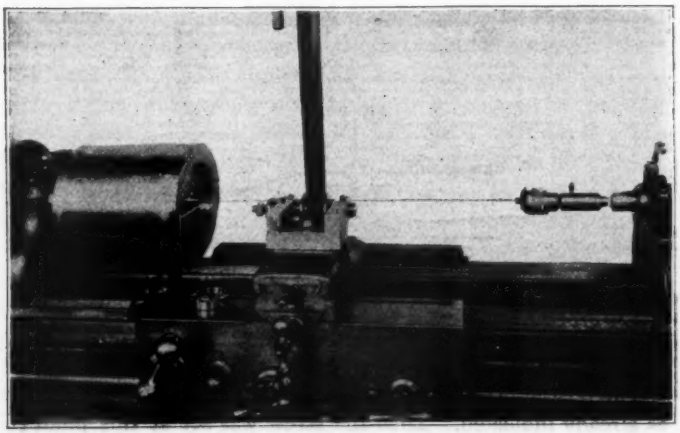


FIG. 5.

When running at 4,400 R.P.M., a notable rise in temperature of the apparatus occurred. At the end of a run of about three hours, a thermometer inserted into one of the test holes rose to 90 degrees F., while the temperature of the metal of the lathe was 58 degrees F. This heating occurred without any metallic contact between plug and ring, as the telephone was frequently used in order to make certain that no contact was occurring.

To determine the friction, the ring was rotated by the lathe spindle, while the rotation of the plug was prevented by the torsion of the $\frac{1}{8}$-inch steel wire, $23\frac{1}{4}$ inches long, the wire being firmly held at the tail stock as shown in Fig. 5. The angle of torsion was read on the vertical scale at the end of a pointer 6 inches long, attached to the plug. The maximum deflection not exceeding $\frac{1}{2}$ inch, the scale reading was taken as the angle of torsion, in-

stead of the sine of that angle. After taking readings for the several speeds, the plug was turned half over to eliminate any possible asymmetry of weight. The results of tests in both positions are given in Fig. 13, together with their mean. The calibration of the wire was effected by freeing the left end from the plug, supporting that end on a horizontal straight edge, and then hanging a small weight on the end of a wooden arm attached horizontally at the end of the wire. The zero reading of the pointer was also taken when the wire was freed from the plug.

The measurements whose results are embodied in Fig. 12 were obtained by measurements of diameters of the plug and the ring, the measurements being made by adapting the micrometer, Fig. 2*c*, first with an extension rod, for measuring the bore, then in a wooden frame, for measuring the plug. The actual form of the surface is, of course, not determined by diametric measurements, but no more satisfactory method of examination was available. The measurements were made for each surface on four diameters, 45 degrees apart, in each of seven sections 1 inch apart along the axis. Four readings were taken on each diameter. The full lines in Fig. 12 show the mean of these measurements for each surface; the broken lines show the maximum measured variation either side of the mean, the radial scale being 1 inch $= \frac{1}{1000}$ inch. The section shown supposes the plug and ring co-axial. It is seen at once that the "fit" of the surfaces is relatively imperfect. It probably would not compare favorably with that of a well worn bearing, such as was used in the experiments of Mr. Tower.

In the curves plotted from the experimental observations, the circled points represent averages of different numbers of observations, as follows:

Fig. 6, two observations.

Fig. 9, 9*a*, 9*b*, 10, two observations.

Fig. 13, two observations.

Pressure curves, two observations.

Thinnest point of film, thirty-two to sixty-four observations.

Fig. 15, six to ten observations.

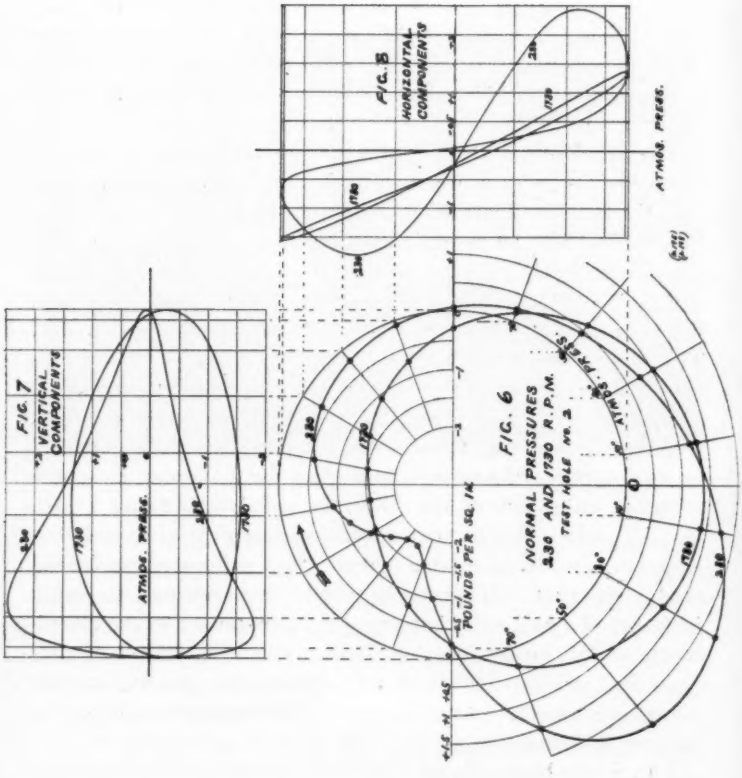


FIG. 5
HORIZONTAL COMPONENTS
ATMOS. PRESS.
FIG. 7
VERTICAL COMPONENTS
ATMOS. PRESS.
FIG. 6
NORMAL PRESSURES
230 AND 1730 R.P.M.
TEST HOLE No. 3
POUNDS PER SQ. IN.
ATMOS. PRESS.

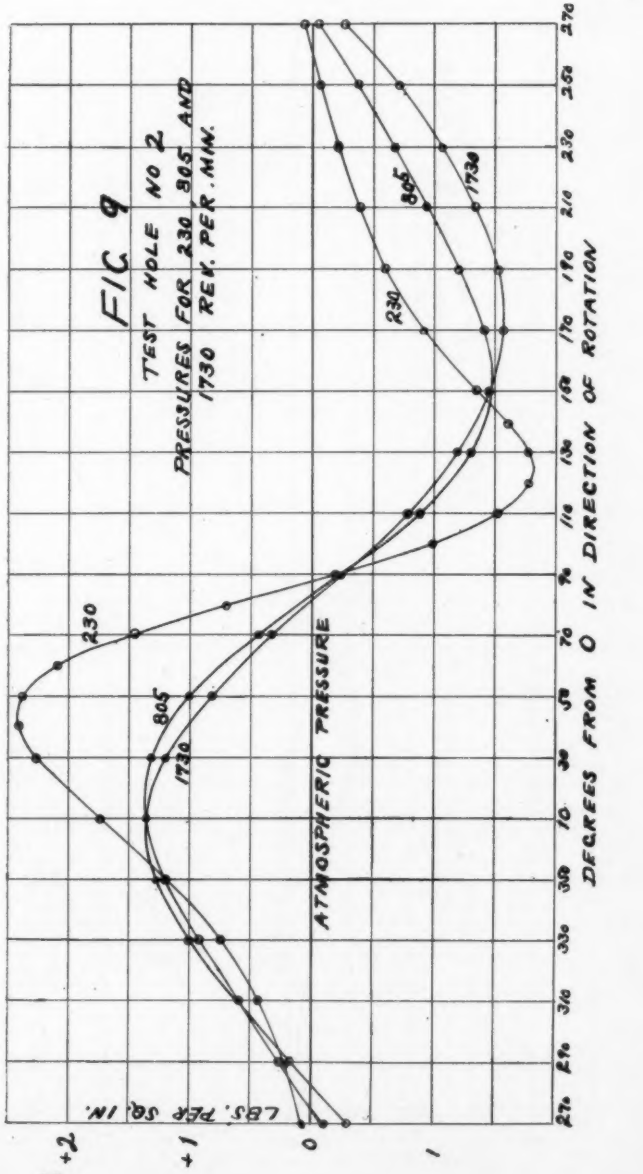


FIG 9
TEST HOLE NO 2
PRESSURES FOR 230, 805 AND
1730 REV. PER. MIN.
230
805
1730
ATMOSPHERIC PRESSURE
LBS. PER SQ. IN.
+2
+1
0
1
DEGREES FROM O IN DIRECTION OF ROTATION
0 10 30 50 70 90 110 130 150 170 190 210 230 250 270
270 290 310 330 350 10 30 50 70 90

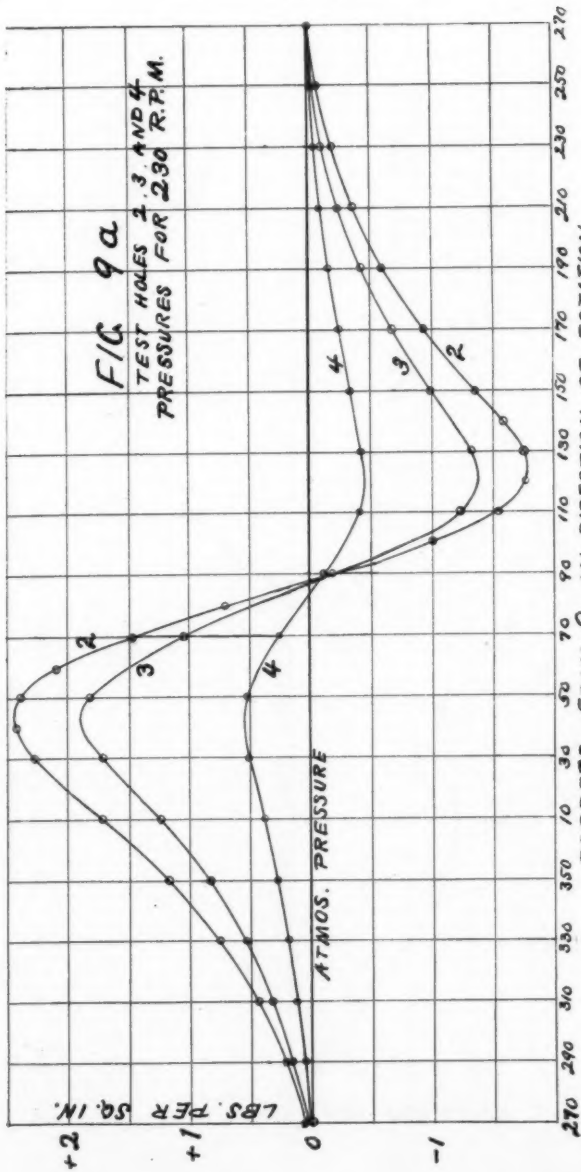


FIG 9a
TEST HOLES 1, 3 AND 4
PRESSURES FOR 230 R.P.M.
LBS. PER SQ. IN.
+2
+1
0
-1
ATMOS. PRESSURE
2
3
4
DEGREES FROM 0 IN DIRECTION OF ROTATION
270 290 310 330 350 10 30 50 70 90 110 130 150 170 190 210 230 250 270

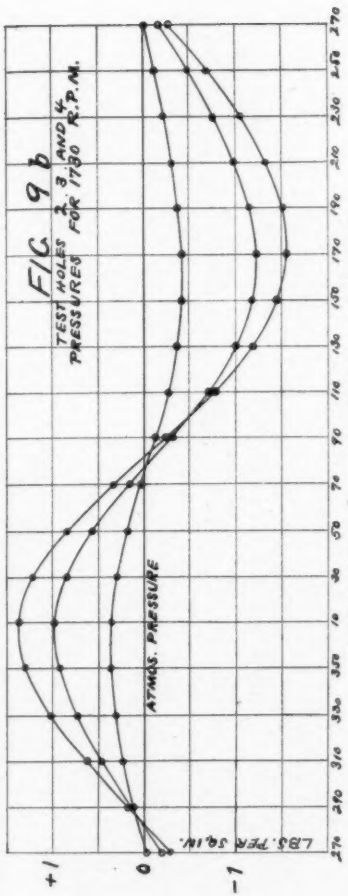


FIG 9 b
TEST HOLES 2, 3 AND 4.
PRESSURES FOR 1730 R.P.M.
ATMOS. PRESSURE
LBS. PER SQ. IN.
+1
0
-1
DEGREES FROM 0 IN DIRECTION OF ROTATION
270 290 310 330 350 10 30 50 70 90 110 130 150 170 190 210 230 250 270

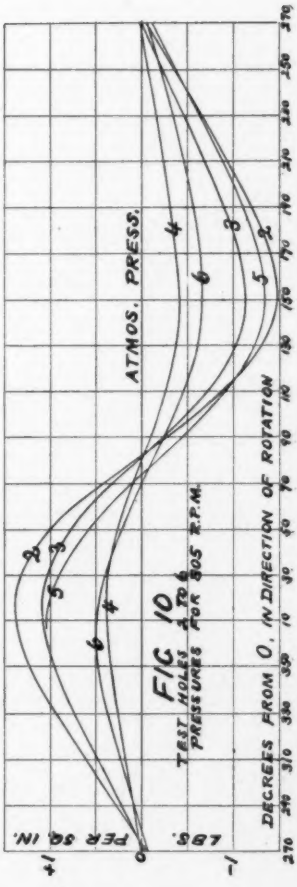


FIG 10
TEST HOLES 1 TO 8
PRESSURES FOR 805 R.P.M.
ATMOS. PRESS.
LBS. PER SQ. IN.
+1
0
-1
DEGREES FROM 0. IN DIRECTION OF ROTATION
270 290 310 330 350 10 30 50 70 90 110 130 150 170 190 210 230 250 270

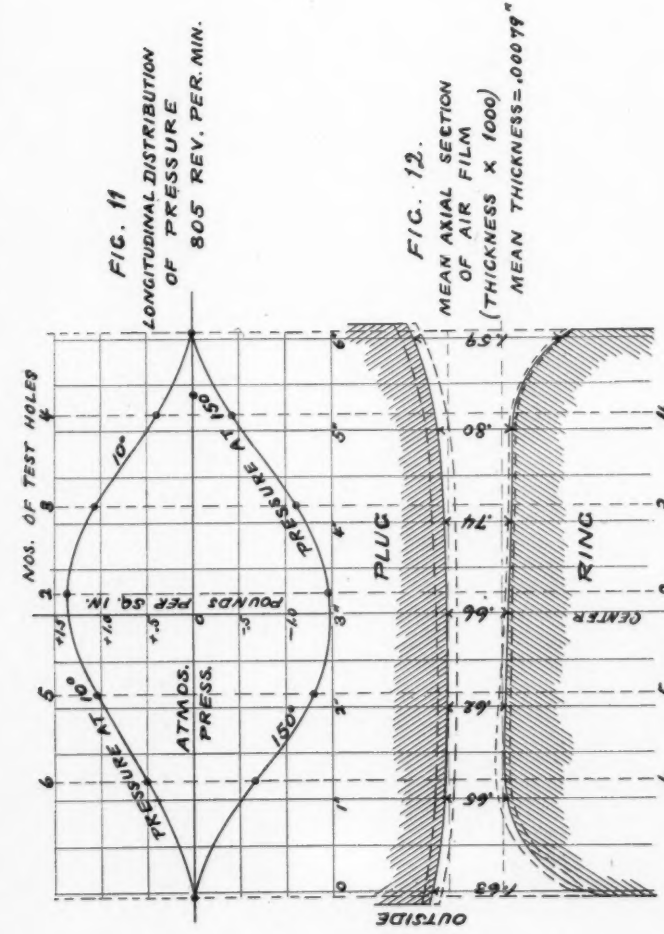


FIG. 11
LONGITUDINAL DISTRIBUTION OF PRESSURE
805 REV. PER. MIN.
NOS. OF TEST HOLES
POUNDS PER SQ. IN.
ATMOS. PRESS.
PRESSURE AT 10°
PRESSURE AT 150°
FIG. 12.
MEAN AXIAL SECTION OF AIR FILM
(THICKNESS x 1000)
MEAN THICKNESS = .00079"
PLUG
RING
CENTER
OUTSIDE
NOS. OF TEST HOLES

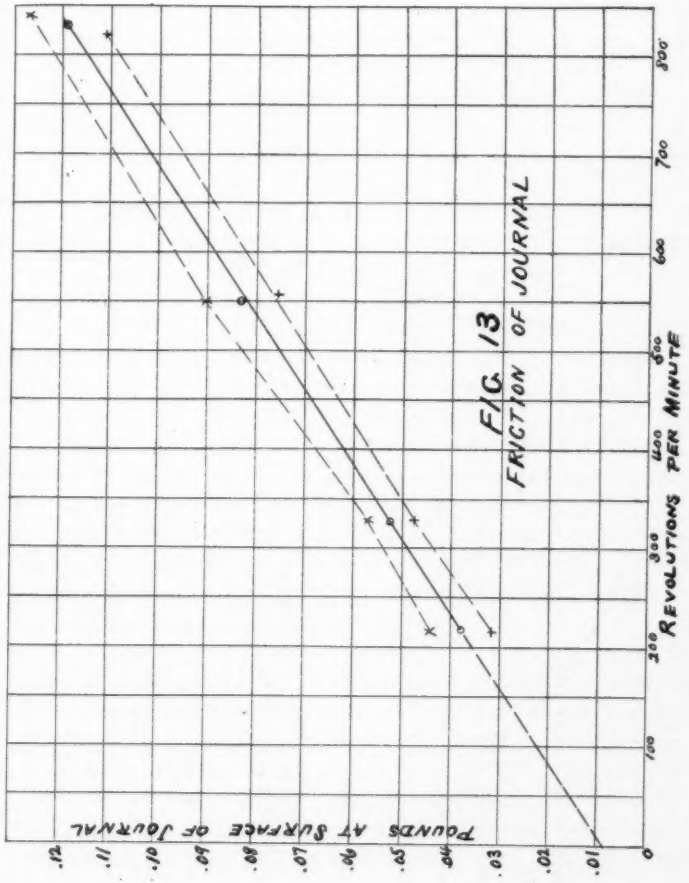


FIG. 13
FRICTION OF JOURNAL

NOS. OF TEST HOLES

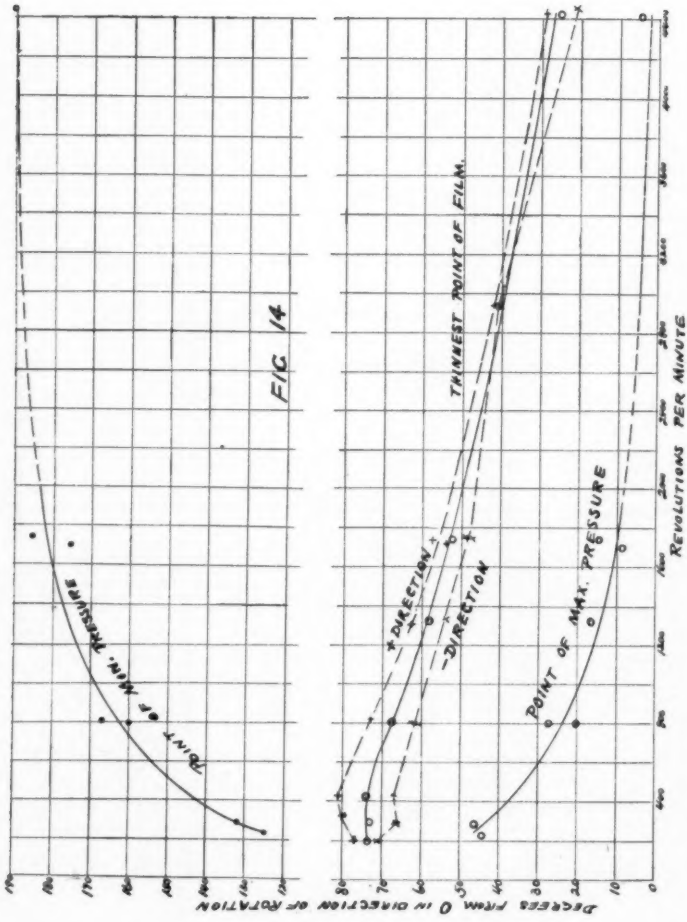


FIG 14
POINT OF MEAN PRESSURE
THINNEST POINT OF FILM.
+ DIRECTION
- DIRECTION
POINT OF MAX. PRESSURE
REVOLUTIONS PER MINUTE
DEGREES FROM 0 IN DIRECTION OF ROTATION

COMPARISON OF RESULTS WITH THE THEORY.

The experiments of Mr. Tower led to the discovery of the separating film of oil and the measurement of the pressures therein. He did not, however, note any variation in the pressure distribution with variation of speed of journal.

Professor Reynolds showed that his theory gave very close agreement with the results of Mr. Tower's experiments, both as to pressure distribution and as to friction of the journal. He also derived a number of interesting conclusions regarding certain quantities in the problem, upon which no direct measurements were taken in Mr. Tower's experiments. Among these conclusions are those referring to the effects of variation of the load and the speed of rotation upon the pressure distribution, the position of the thinnest point of the film and the distance between the axes of brass and journal. In the present experiments, direct measurements were made upon these quantities, the results appearing to correspond with the general statements derived from the theory. The theory was derived for perfectly cylindrical surfaces and for an incompressible fluid lubricant, of constant co-efficient of viscosity at any given temperature, while in these experiments the surfaces are far from perfect; the fluid is highly elastic, and of a co-efficient of viscosity which varies with the variations of pressure and consequent changes of temperature; hence results of these experiments could not be expected to conform exactly to the theory.

The conclusions referred to which bear directly upon the present experiments may be summarized as below. Excepting the first and second, they are derived from equations of general applicability within the limitations just noted.

References are to Fig. 16, which is here reproduced from Professor Reynolds' paper. It represents a right section of a cylindrical journal and its brass; the rotation of the journal is counter-clockwise. The conditions are such that the film is complete.

1. The vertical components of the pressure and friction balance the load.

2. The horizontal component of the oil pressure balances the horizontal component of the friction.

L
G
Q
H
O
P
F
C
A
J
B
D
I

Fig. 16.

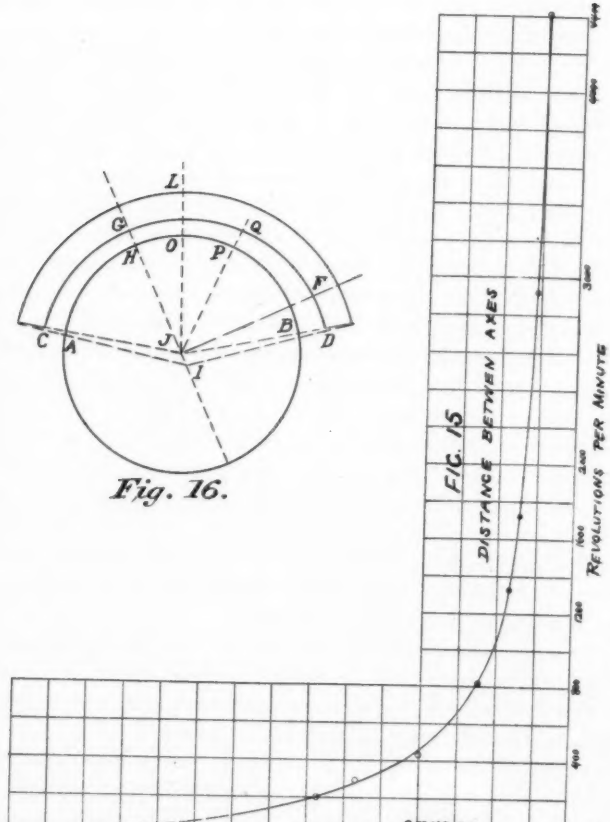


FIG. 15
DISTANCE BETWEEN AXES
REVOLUTIONS PER MINUTE
INCHES
.0007
.0006
.0005
.0004
.0003
.0002
.0001
0
400
800
1200
1600
2000
3000
4000

3. When the brass is unloaded, its point of nearest approach will be its middle point. The pressures are then symmetrically disposed with reference to O, the positive on the right, the negative on the left.

4. As the load increases, the positive vertical component must overbalance the negative component. This requires that H should be at the left of O.

5. As the load increases, OH reaches a maximum value which places H nearly but not quite at the left extremity of the brass, but still leaves JI small as compared with GH. For a further increase of the load, H moves back again toward O.

6. As the load increases, the distance between the axes of brass and journal increases.

We may now examine the results of the present experiments in the light of these conclusions.

First. By constructing curves similar to those of Fig. 7 for test holes No. 3 and No. 4 in addition to those for No. 2, it has been found that the resultant upward pressure in the film for the corresponding half of the film is 21.1 pounds at 230 R.P.M. and 22.6 pounds at 1,730 R.P.M.

Assuming (as did Mr. Tower in his experiments) that the resultant on the other half would be the same, this gives totals of 42.2 and 45.2 pounds respectively, while the actual load is known to be 50.5 pounds. The friction, being relatively very small (see Fig. 13), is here neglected. A possible explanation of this discrepancy of 5.3 to 8.3 pounds in a total of 50.5 may be found in the method of testing. As the ring is turned on its bearings, each new position involves a new distribution of pressures, since the bore is not perfectly cylindrical; and the test hole gives the pressure at its position in the new distribution. Thus it is improbable that the curves should represent exactly the distribution existing for any one position of the ring.

The derivation of the curves of Figs. 7 and 8, from those of Fig. 6, is obvious.

It will be noted that the resultant upward pressure for either curve (Fig. 7) lies to the left of the central vertical. This pressure then forms with the load (acting downward through the center) a couple. To balance this couple, we find in Fig. 8 an

opposed couple, indicated by the positive and negative loops of the horizontal pressure. These loops are approximately equal and opposite, thus nearly fulfilling the second of the conditions given. The friction is here again neglected.

Conclusions 3 to 6 contain references to varying load. In these experiments the load was constant, while the speed was varied.

By examining the equations from which the conclusions were derived, it is found that the load and the speed are inversely related; that an increase of load has an effect upon the form of the film of the same kind as that of a decrease of speed, and *vice versa*. For the third conclusion, the pressure distribution could not apply to the present case, owing to the load of 50½ pounds always being present. However, by referring to Fig. 14 it will be seen that the point of nearest approach is nearer *O* as the speed increases, and if it were possible to make the ratio speed ÷ load approach infinity, the thinnest point of the film would probably approach *O* indefinitely.

In the fourth conclusion, the reference to the pressure distribution again is not strictly applicable to the present case; but by referring to Fig. 14, it is seen that the point of nearest approach is always on the "off" side of *O*.

Fifth. In Fig. 14 it seems that as the speed decreases, the point of nearest approach recedes from *O* to a distance of about 74 degrees at a speed of 200 R.P.M. The curve from the experiments seems to indicate at 300 R.P.M. the maximum recession of *H* from *O* as predicted by Professor Reynolds. Unfortunately the irregularities of the surfaces would not permit reduction of the speed below 200 R.P.M. without contact taking place at some points, as shown by the telephone; indeed it was only with difficulty that it was made to run at the speed of 200 revolutions per minute. The distance between axes *JI* is at the speed 300 about .00027 inch (see Fig. 15), while the distance $GH = .00079$ inch (mean difference of radii) $- .00027$ inch $= .00052$ inch. Thus *JI* is about one-half *GH* when *H* is farthest from *O*.

The sixth conclusion is verified as shown in Fig. 15, again remembering that decrease of load corresponds in effect to increase of speed.

THE DESIGN AND TESTING OF CENTRIFUGAL FANS.

By Hammersley Heenan, M. Inst. C. E., and William Gilbert, Wh. Sc. Assoc. M. Inst. C. E.

[Read before the Institution of Civil Engineers, London.]

The experiments recorded in the paper have recently been made by the authors upon several of the best known varieties of fans in use, and had for their object:

(1) To determine the best shape of the fan blade and fan case, in order to secure a minimum expenditure of power when producing any given output of air, *i. e.*, to find the best type of fan;

(2) The standard type being selected, to obtain data whereby the proper diameter of the standard fan and its most economical speed could be determined for any required output of air at a given pressure.

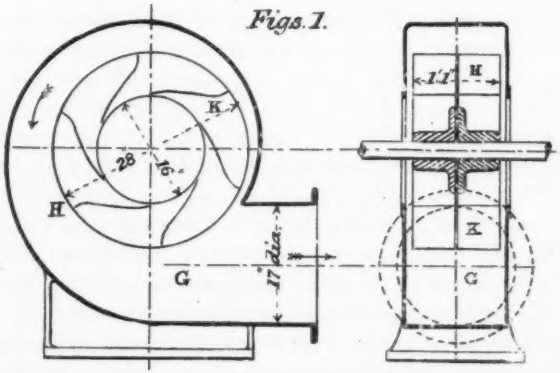
The experiments proved that a fan with a few simple blades gives the best result, provided the form of blade and the dimensions of the casing are designed to suit the kind of work required. Fans of more complex design have too large an internal resistance to give the highest mechanical efficiency, although they may have to be used if high pressures are essential. In the study of air propellers there is great difficulty in taking correct measurements of the pressure and quantity of the air delivered, if the velocity of the stream is at all great. From this cause, experiments, apparently conducted with the greatest care, have often shown a fan under test to give an efficiency of over 100 per cent., and therefore, general statements of the efficiency of fans should be received with great caution. Considerable doubt also appears to exist as to the proper method of comparing the performance of different fans. To meet the first difficulty, the authors carefully tested and re-arranged the measuring instruments until

results known to be consistent were obtained. Particulars of these tests will be given later in the paper, and the authors now proceed to give an outline of the manner in which the performance of any fan should be tested and put on record.

Preliminary.

A common type of blast fan, Figs. 1, consists of a center or drum *HK* fitted with, say, six blades, revolving within a casing. The air enters through the center of the sides of the drum, and is discharged by the outlet *G*. Suppose such a fan to be running at a constant speed, taking air from the atmosphere and discharging through a delivery tube *HK*, Figs. 2 in the Plate, having an outlet at *K*, the area of which can be varied. Let the fan center

Figs. 1.



be assumed to be 28 inches and the delivery tube 17 inches in diameter. Let there be two water gauges, *A* and *B*, one of the branches of each of which is connected to a pipe soldered into the side of the delivery tube in the gauge *A*; the other branch is attached to a pipe passing to the center of the delivery tube, the end being suitably bent to face the stream. The pressure in the pipe *C* (inductive action being prevented) will be that due to the compression of the air only, but the pressure in the pipe *D* will be that due to pressure and velocity combined. Hence, the gauge, *A*, which indicates the difference between the pressures

in the pipes *C* and *D*, will record the pressure due to velocity only. The compression of the air can be measured by the second gauge *B*.

First, if the outlet be closed so that no air is delivered, the gauge *A* will remain at zero, since there is no flow of air through the tube, but the second gauge *B* will indicate a considerable compression, about 11½ inches of water, if the tip speed of the fan (supposed to remain constant throughout the experiment) is 12,000 feet per minute. Next, let the end of the delivery tube be opened to give, say, half the area of the outlet *K*. The fan now passes a considerable quantity of air, about 8,000 cubic feet per minute, and, on account of this flow, the velocity gauge *A* will indicate nearly 1¾ inches of water. The compression, as shown by the gauge *B*, will have fallen to 8 inches. During the time the outlet of the delivery tube was closed, with no air being delivered, the efficiency was of course 0. But when the fan is passing 8,000 cubic feet of air, under a compression of 8 inches of water, the efficiency reckoned on the compression alone will be about 66 per cent., 15 H.P. being required to drive the fan. When the outlet of the delivery tube is fully opened, the fan delivers freely to the atmosphere, and gauge *B* shows that the air is under no compression whatever. But the amount of air discharged has increased to about 13,700 cubic feet per minute, and the passage of this quantity of air through the delivery tube shows a velocity on the gauge *A* of nearly 5 inches of water. Since the air is not compressed but merely expelled at atmospheric pressure, the efficiency now reckoned on the compression is 0. The compression alone enables the fan to do useful work in overcoming resistance. The energy due to velocity will be almost always wasted in shock.* Since then the efficiency for no delivery, or for maximum delivery, is zero, there is necessarily an intermediate delivery at this particular tip speed of 12,000 feet per minute, for which the efficiency, reckoned on the compression

* It may be as well to point out that there are cases where the velocity energy may be as important as the compressive, for instance, where large quantities of air have to be exhausted from a building. Here both gauge readings would probably be exceedingly small.

produced on the air, is a maximum. This may be exhibited, together with the other results of the preceding experiment, as follows:

Take two axes, *Ox Oy*, at right angles, Fig. 3 in the Plate, and for any size of opening at the end of the delivery tube set off along *Ox* the volume delivered by the fan, as *Oa* for the half area of the opening mentioned, when the delivery was 8,000 cubic feet per minute. On the ordinary *ac* set up *ad* (8 inches), the compression produced at that delivery, *ae* to represent the brake H.P. required to drive the fan, and *ag* the calculated efficiency. Mark off other values of the compression, brake H.P., and efficiency at *O*, where the delivery is zero, at *b* where the delivery is 13,700 cubic feet per minute, and for any other intermediate points where readings have been taken. The curves shown in the figure may then be drawn. It will be seen that the compression curve *hb* is remarkably like the external characteristic curve of a separately excited dynamo, if for volume be read current and for compression electromotive force; whilst the variable opening at the end of the delivery tube corresponds with the resistance of the external circuit. The internal resistance of the fan represents the armature resistance. The point *b* is equivalent to the dynamo short-circuited.

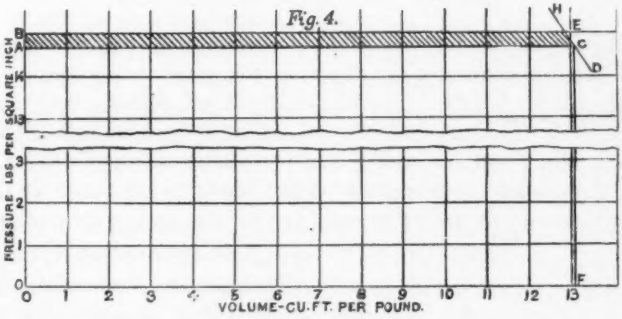
These curves, for any tip speed, may be called the characteristic curves of the fan for that speed. If the curves are drawn for tip speeds of 5,000 feet, 6,000 feet, etc., up to 14,000 feet per minute, they form a complete picture of the capabilities of the fan.

Referring to Fig. 3 in the Plate, it will be seen that the H.P. required to drive the fan increases with the area of the outlet. The use of a small outlet restricts the delivery of the fan, and hence the H.P. required is less.

To plot the efficiency curve on Fig 3, the useful work done by the fan must be known. This, for any given values of the compression and discharge, has been estimated as follows:— Take the case of a fan drawing air from the atmosphere and delivering into a reservoir under pressure. Let *OF* and *FC*, Fig. 4, represent the original volume in cubic feet, and the pressure

in pounds per square foot, of 1 pound of the air which enters the fan, and let AB represent the pressure, by water gauge, produced by the fan. The air will be compressed along the adiabatic DCE and expelled from the fan into the reservoir along the line ED, the shaded area $BACE$ being the diagram of work done. The diagram is drawn for a pressure, AB, of 10 inches of water, which is seldom reached in large fans; therefore, for

Fig. 4.



all practical purposes the work done may be taken as the rectangle AC, AB, the corner cut off by the compression curve being neglected. Hence, the work done by the fan, in foot pounds, per pound of air entering the inlet is:

Volume of air in cubic feet per pound $\times$ increase of pressure in pounds per square foot, and where,

v = volume of air entering the fan in cubic feet per minute,

i = pressure, or compression, measured in inches of water, a pressure of 1 inch of water being equivalent to 5.2 pounds per square foot,

$$\text{the horse power of the fan} = \frac{V \times 5.2\, i}{33{,}000} = \frac{V \times i}{6{,}352} \quad . \quad . \quad (1)$$

It appears reasonable to assume that the compression curve is adiabatic, for since the air passes through the fan in a very small fraction of a second, there would be no time for the abstraction of heat by the casing. However, the exact class of this curve does not much affect the calculation. The efficiency of the fan

is taken to be its effective work, divided by the power expended in driving the fan shaft for a given time. For an increase of pressure of 10 inches of water there is a rise of temperature of $3\frac{3}{4}$ degrees F. due to the adiabatic compression. If the conditions, however, are such that the air regains its original temperature before being used, the volume will diminish about $\frac{3}{4}$ per cent. and the effective work of the fan be correspondingly reduced.

VARIATION OF THE SHAPE OF THE BLADES.

Having established a method of representing the performance of a fan, the change in the characteristic curves when the shape of the blades is varied may be examined. Suppose a fan of the type shown in Figs. 1 be selected, and in succession with blades No. 2, No. 3 and No. 4, Fig. 5. The angles which the edges of the three blades with a plane perpendicular to the radius at the point considered, are 25 degrees at the inlet and 35 degrees, 60

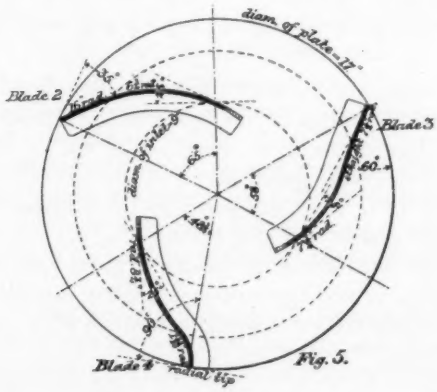


Fig. 5.

degrees and 90 degrees, respectively, at the outlet, the fan center being 17 inches in diameter and 8 inches wide. As far as possible in each case the air passes on to the blade without shock. The angle between the blade and the circumference of the inlet circle is obtained by compounding the radial velocity of the flow of the air through the drum, with the tangential motion of the

air relatively to a point in the circumference of the inlet circle. This angle would not be the same for all speeds and discharges, but a slight variation of the required angle is of little importance. Figs. 6, 7 and 8 in the Plate, give the characteristic curves of these blades for a constant tip speed of 12,000 feet per minute. The "total gauge" in the figures is the gauge reading due to pressure and velocity combined, from which the total efficiency is derived.

It will be seen that the radial tip blade No. 4 gives the best result. Blade No. 3, with a tip angle of 60°, is not greatly inferior; but blade No. 2, having a tip angle of only 35°, gives a low efficiency, owing to the rapid drop of the compression curve as the discharge of air increases.

EXPERIMENTAL APPARATUS.

The arrangement of the apparatus whereby the actual measurements of the brake H.P., volume of air discharged, compression and speeds were obtained in the case of fans of moderate dimensions is shown in Figs. 9 in the Plate. The fan is driven from the countershafting *BB*, which derives its motion from a spherical steam engine *C*. The outlet of the fan is connected by short circular iron delivery tube with a boiler flue *EE*, 2 feet 6 inches in diameter and 18 feet long. At the center of the flue a partition *F* is fitted, to which can be attached a series of plates having circular orifices of various sizes, varying between $4\frac{3}{4}$ and 18 inches in diameter. Each orifice in turn acts as a constant resistance to the fan. A well-cut outlet of known diameter is placed at the end of the flue *G*, where the velocity of the air was measured by an anemometer. This opening was made much larger in diameter than the outlet of the fan, in order to avoid injuring the anemometer by a violent current of air.

This outlet and the levers whereby the anemometer was moved over all portions of the outlet, the instrument being kept truly perpendicular to the axis of the flue, are shown in the end elevation. Very consistent air readings were obtained in this manner. The pulley *H* driving the fan, was not keyed to the shaft, but was driven by it through an Emerson power scale, a form of

transmission dynamometer in which the moment of the driving effort is balanced, through a system of levers, by a pendulum moving over a graduated scale. A speed counter records the revolutions of the shaft. This instrument was tested by the authors at the conclusion of the experiments by applying a band brake to the pulley *H*, and on comparing the two readings the difference was almost inappreciable. A tachometer *K*, coupled to the countershafting *BB* by a spiral spring, enabled the speed of that shafting to be regulated and the proper tip speed to be approximately maintained by the man in charge of the engine. A small hand counter, carried in a sliding frame *L* and applied when necessary to the fan spindle, enabled the exact number of revolutions per minute of the fan to be obtained.

Measurements of the pressure and the velocity of the air stream were taken at the section *MM*, of the delivery tube. The velocity varied greatly in different positions on the same cross section of the tube. Readings were taken at several points in a cross section by means of two gauges, each of which could be traversed over a diameter at right angles to the other. It was afterwards found unnecessary to measure the velocity, since it can be readily calculated when the discharge, as measured by the anemometer, is known; but at the same time the velocity, as measured by the velocity gauge, gives a check on the anemometer readings.

If the cross section of the delivery tube be divided into a number of imaginary areas, and the square root of the mean gauge reading for each area be multiplied by that area and by a suitable constant, then these results added together give the total discharge of the fan.

Table I illustrates the variation of velocity referred to; it gives the gauge reading due to velocity in four positions along a diameter of the cross section of the delivery tube where the gauge was fitted, as taken simultaneously.

The measurement of the compression presented some difficulty, owing to the fact that the air flowing across the end of the side gauge, caused a large amount of induction; a vacuum being often recorded where a pressure was known to exist. The

authors are indebted to Professor W. C. Unwin, F.R.S., for the information that a plate placed across at the end of the tube would prevent this inductive action. The form of side gauge used is shown in Fig. 28 in the Plate. It was tested by the authors in a manner subsequently described, and was found to give very correct results.

TABLE I.

Position.	Gauge reading due to velocity.	Air velocity.	Distance from center of tube.	Remarks.
	Inches.	*Ft. per second.*	*Inches.*	
1	3.20	118	+ 3.1	Diameter of delivery tube, 8.9 inches.
2	2.88	112	+ 1.1	
3	1.85	90	− 1.1	
4	1.70	86	− 3.1	

To draw the characteristic curves from the experimental results, seven resistance plates, as before mentioned, lettered *A B C D E F*, with graded circular orifices, were arranged to fasten on to the center *F* of the boiler flue. Two observations were taken with each plate of the discharge, the compression and the horse power supplied to the fan at, or near, each of the tip speeds 5,000 feet, 6,000 feet and 12,000 feet per minute. Owing to the slip of the belt on the fan pulley at high speeds, and from other causes, it was impossible to maintain the tip speed quite constant for any length of time; hence the following laws were assumed, and were afterwards proved to be true:

(1.) The air discharge varies as the speed
(2.) The gauge reading varies as the $(\text{speed})^2$
(3.) The brake H.P. varies as the $(\text{speed})^3$
} for a constant resistance.

The results were then plotted thus:—On a sheet of sectional paper the square root of the compression, for any given plate, was plotted, with the speed recorded by the counter. Since the compression for a constant resistance varies as the square of the speed, the points fall approximately on a straight line. These lines were obtained on the same sheet for each one of the plates *A B C D E F*, and from them for any intermediate speed the corresponding square root of the compression could be obtained.

In the same way the speed was plotted with the air discharge, and with the cube root of the horse power given to the fan.

Using these curves of water gauge, air discharge and brake H.P., characteristic curves were drawn for tip speeds of 5,000 feet, 6,000 feet, 7,000 feet, to 12,000 feet per minute. The method is as follows (see Fig. 3 in the Plate):—Suppose the tip speed be 12,000 feet per minute. Taking two axes *Ox*, *Oy* at right angles and using *Ox* as a volume scale, set out *O*1, *O*2, *O*3 . . . *O*6 to represent the volume passed through the plates *A*, *B*, *C* . . . *F* at this constant tip speed, taking the data from the air discharge curve. Draw the ordinates 1*A*, 2*B*, &c., and mark off, on the ordinate representing each resistance plate, the compression and the brake H.P. taken from the interpolation curves previously mentioned. From the curves plotted, points on the efficiency curve can be calculated and the curve may be drawn on. The gauge reading due to velocity is calculated, and added, as also the total efficiency, that is, the efficiency reckoned on the gauge reading due to velocity and pressure combined. From the method of obtaining the curve, it will be seen that if *OA* represents any volume delivered by the fan, then *ad*, will be the corresponding compression, *ae* the brake H.P., and *ad* the efficiency for that volume and pressure. The point *g* might be taken as the best working point on the efficiency curve, so that the best output for this fan, at the tip speed of 12,000 feet per minute, is 8,000 cubic feet at a compression of 8 inches of water. A comparison of the characteristic curves of two different types of fans, at the same tip speed, gives an excellent idea of their relative merit.

All the tests recorded of Heenan fans were made on centers with parallel sides, and with the proper clearance in the case to allow the air to flow uniformly from the fan center at all portions of its circumference.

With the apparatus described, tests were made on several fans of various types. Fig. 10 shows the characteristics obtained at a tip speed of 12,000 feet per minute from a fan 16 inches in diameter, having tapering side plates. The diameter of the inlet was $5\frac{7}{8}$ inches, the widths of the fan center at the inlet and outlet

being $7\frac{1}{4}$ and $1\frac{1}{8}$ inches respectively. This fan ran in a concentric case, the clearance being $1\frac{1}{8}$ inches. The compression obtained with the outlet closed was 9.4 inches against 11.2 inches of blade No. 4, Fig. 5. Further, the compression obtained fell off very rapidly with the increase of output, so that the working compression would not be over 8 inches. The efficiency was also low. Fig. 11 shows the characteristic obtained at 12,000 feet per minute from a fan center 12 inches in diameter, the maximum width being $2\frac{3}{4}$ inches. The center rotated in a whirlpool chamber of 23 inches diameter. The center was of cast iron and the tip angle of the blades was 30 degrees. Of course, the efficiency measured would not be representative in so small a fan, but the maximum compression did not exceed 4.4 inches. The effect of a whirlpool chamber is to produce a very quiet fan.

TEST OF A MINE VENTILATING FAN.

Of the tests of mine ventilating fans made by the authors, that of a fan supplied to the Parkend Colliery Company, South Wales, is selected for illustration, as giving a fair average result. The managing director of the company, Mr. F. H. Deakin, Assoc. M. Inst. C. E., placed the fan at the authors' disposal for this test. The fan, in connection with the approach tunnel and ventilating shaft of the mine, is shown at Figs. 12. The fan center is 7 feet in diameter and 2 feet wide. The upper por-

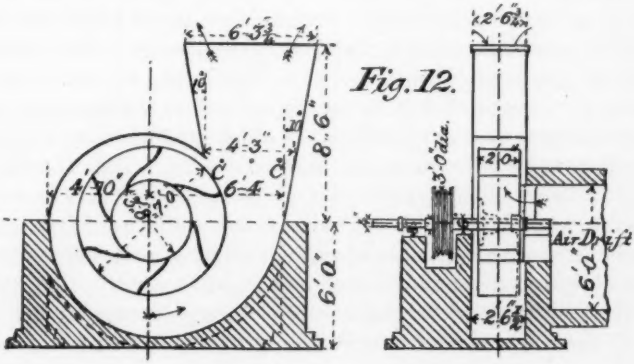


Fig. 12.

tion of the case and evasee chimney is built up of wrought iron plates, the lower portion being formed in brickwork $4\frac{1}{2}$ inches thick. The fan was driven by a horizontal non-condensing engine, the cylinder being $12\frac{1}{8}$ inches in diameter and of $17\frac{3}{4}$ inches stroke. Cotton ropes were used for driving, two only being in operation at the time of the experiment. The engine was an old one, and was not supplied with the fan. The boiler pressure averaged 40 pounds per square inch.

To provide a variable resistance for the fan, three 9-inch by 3-inch planks were placed across the mouth of the air drift, and boards nailed to these planks restricted the flow of air to the fan more or less as required. The folding doors at the top of the ventilating shaft were open during the whole of the test. A tachometer driven by a belt from the fan shaft enabled the approximate speed of the fan to be judged and regulated by the man in charge of the engine; the number of revolutions in a two-minute reading being obtained by means of a hand counter held to the fan shaft. The engine speed was obtained by a small counter applied to the shaft in the same manner. The air discharged was measured by an anemometer held at the top of the fan chimney, a staging being erected for that purpose. The area of the top of the chimney was divided into eight equal rectangles by means of wires tightly stretched across, and the anemometer, attached to a small iron tube, was held for a quarter of a minute in each division. In this fan the flow of air was fairly uniform over the whole of the outlet area of the chimney, but in some cases, where the fans were run slowly for experimental purposes, guide vanes had to be fitted in the base of the chimney to secure the result mentioned. Four degrees of opening were arranged at the adjustable orifice, and for each of these readings were taken with tip speeds of 4, 5, 6, 7 and 8,000 feet per minute. The duration of each reading was two minutes, and it was taken twice. The observer with the watch was stationed in the engine room, and signalled by an electric bell to the observers at the anemometer and the fan counter. Indicator diagrams were taken from the ends of the engine cylinder during each reading. The vacuum produced by the fan was measured

by a side gauge, placed in the air drift close to the fan inlet, and a pipe led from this tip to a water gauge placed on a table outside. Three side gauges, *CC*, Figs. 12, were also placed at the root of the chimney, so that the vacuum produced, and, consequently, the efficiency of the chimney, could be determined. It was found that the vacuum was practically the same at each of the three gauges, so that only one of them was read.

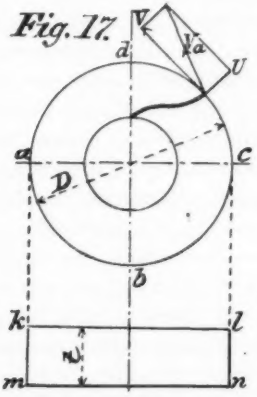
Figs. 13 to 16 in the Plate give the characteristic curves obtained from this test. The dotted lines *ee* correspond to the resistance offered when the choking boards were removed and the fan took air from the mine only. The maximum efficiency increases with the speed, and is 67.2 per cent. at a tip speed of 8,000 feet per minute, and 70.3 per cent. at 9,000 feet per minute. The fan was designed to pass 20,000 cubic feet of air per minute, which it was estimated would require a $3\frac{1}{2}$-inch head, as measured by the water gauge, to force through the mine, the speed being 300 revolutions per minute. Referring to the ordinate *ee*, which represents the mine on the characteristic curve for 7,000 feet tip speed, or 318 revolutions per minute, the compression is 3.45 inches of water, and the discharge per square inch of center section is 11.6 cubic feet, corresponding to a total discharge of 23,150 cubic feet per minute, so that the fan is amply large for the work.

COMPARATIVE OUTPUT OF DIFFERENT FANS.

Several fans of the same type but of different sizes having been tested, the dimensions of the fan to which the output is proportional can be ascertained. Suppose two fans of different dimensions, but running at the same tip speed, are fitted with delivery tubes, as in Fig. 2, and let the area of the resistance plates be so adjusted that the velocity of the air through each fan center is the same. The air velocities in the respective fan centers may be conveniently compared by dividing the discharge of each fan by the circumferential area of the fan center, that is, by the area of a rectangle the adjacent sides of which are the circumference *a b c d*, and the width *t*, Fig. 17. This would give the radial component of the velocity of discharge, *V*, from each fan. Since the fans are of the same type, and run at the same tip speed, the

total pressure generated by each fan will be the same. It will be greater than the total water gauge in each delivery tube by the amount necessary to overcome the resistance of the fan itself to the passage of the air through it. The loss of pressure in the fan will be partly due to skin friction, but in a far greater degree to shocks and eddies caused by abrupt changes in the velocity and direction of flow. The velocity of the air being the same at corresponding points in each fan, the pressure lost in each fan will be the same, since the number of abrupt changes was the

Fig. 17.



same. The available pressure in each delivery tube will therefore be the same. It may therefore be concluded that if two fans of different dimensions but of a similar type are running at the same tip speed and give the same pressure, the velocity of the air through each fan center will be the same; and hence the volume of air discharged per minute by each fan will be proportional to the circumferential area of its revolving center, or more conveniently to the area of the rectangle *k l m n*, Fig. 17, formed by the diameter and width of the fan. This may be termed the center section. The volume discharged per unit of area of center section should be the same for all fans of the same type, if the tip speed and pressure are the same. Hence in plotting the characteristics for different fans, if, instead of the actual volume

delivered, there be plotted the volume discharged per square inch of center section of the fan in question, the characteristics should be identical.

This is the manner in which the characteristic curves were usually plotted, and Figs. 18, 19 and 20 in the Plate show them for 18½-inch, 28-inch and 48-inch Heenan fans, at a tip speed of 6,000 feet per minute, the center section areas being 171, 364 and 726 square inches respectively. The brake horse power is not divided by the center sectional area, but the efficiency is known, and the brake horse power can be readily calculated from the data. From inspection of the three characteristics just referred to, it will be seen that the working output of these fans, at a tip speed of 6,000 feet per minute, might be taken at 12 cubic feet per square inch of center section per minute. The compression would be 2 inches of water, and the efficiency at this delivery, reckoned on the compression alone, 56 per cent. To save weight and space, the working point is selected where the output is 10 per cent. to 15 per cent. greater than at the point of maximum efficiency. Referring to Fig. 19, it will be seen that the maximum efficiencies of the 28-inch fan are 73 per cent. and 80.5 per cent. respectively, reckoned on the compression and the total gauge readings.

DIMENSIONS OF A FAN FOR A GIVEN OUTPUT.

Supposing a set of the characteristics mentioned to be drawn for each tip speed between 5,000 feet and 14,000 feet per minute; then if a fan be required to give any number of cubic feet of air per minute at a given reading of the water gauge, its dimensions can be obtained as follows. Find for what tip speed the required compression is obtained near the point of maximum efficiency. This speed being selected and the working point on the efficiency curve fixed, the efficiency and volume per square inch of center section are known. This fixes the brake horse power and also the area of the center section, since the total delivery is given. A suitable ratio is maintained between the diameter and the width of the fan center, and the revolutions are fixed to give the required tip speed. In designing a fan, it would usually be

arranged to work at a point of the efficiency curve corresponding to *g*, Fig. 3 in the Plate; hence the fan would have a working output of rather more than half the volume of air it would discharge if free open to the atmosphere. It may be noticed that measurements of the water gauge reading and the volume, etc., of a fan when discharging freely into the atmosphere, are not of much assistance in determining the real value or efficiency of the fan, although such figures are often given.

Since the fans are usually designed to work at one point on the characteristic curve, it would obviously be convenient to have a single set of curves which would give the information required about this working point for all tip speeds. These curves may be obtained as follows. Let an ordinate be ruled on the characteristic curve, at the best working point, for several of the tip speeds between 5,000 and 14,000 cubic feet per minute. Tip speeds are then plotted as abscissas, with the compression and volume per square inch of center section as ordinate, the value of the efficiency reckoned on the compression only being written on the volume curve. A set of such curves for the design of ship fans are shown in Fig. 21 in the Plate. The working point is not the point of maximum efficiency, but a point where the output is 10 to 15 per cent. greater than that.

As an example of their use, suppose a fan is required to pass 9,000 cubic feet of air per minute at 4 inches on the water gauge. Referring to the figure, it will be seen that *b* is the point on the compression curve where the compression is 4 inches. Drawing an ordinate at this point, cutting the volume curve at *c* and the base line at *d*, the volume passed per square inch of center section is seen to be 14.6 cubic feet per minute, and the corresponding tip speed, 7,900 feet per minute.

The required area of center section is:

$$\frac{9{,}000}{14.6} = 616 \text{ square inches.}$$

Assuming the ratio of width to diameter to be $w : 0.18\ d$,

$$0.18\ d^2 = 616;$$

hence $d = 58.6$ inches, and $w = 10\frac{1}{2}$ inches.

A fan 60 inches by 11 inches would be used, and the correct revolutions are—

$$\frac{7,900}{5\,\pi} = 502 \text{ per minute.}$$

The efficiency taken from the curve is 56.5 per cent.; hence the I.H.P. of the engine would be—

$$\frac{9,000 \times 4}{6,352 \times 0.565} = 10.0 \text{ I.H.P.}$$

The efficiency, 56.5 per cent., is reckoned on the compression only. The total efficiency as obtained from the characteristic curve is here 66 per cent.

TEST OF WATER GAUGE TIPS.

The apparatus by means of which the accuracy of the tips used for the measurement of air pressure and velocity was tested is shown in Figs. 22 in the Plate. To test the facing gauges, the following method was adopted: The tip was moved at a known velocity through air at rest, and provision was made whereby a water gauge recorded the pressure thus set up. *BB* is a wrought iron circular tank, 7 feet in diameter, and divided into eight compartments by vertical dash boards *C*. The tip to be tested was screwed into a horizontal pipe, *DE*, attached to and rotated by the hollow vertical spindle of the apparatus. The top bearing of this vertical spindle was formed in a hardwood block, a stuffing box being used in connection with it. From the upper portion of the block a pipe led to the water gauge. Free communication was thus established between the tip under test and the recording water gauge. The vertical spindle was driven from a counter shaft by a belt pulley and suitable gearing, the revolutions per minute being recorded by the hand counter *F*. The tip under test was arranged to describe a circle 20 feet in circumference.

The object of the dash boards being to prevent motion of the air in the tank, the openings in them through which the tip and pipe passed were made as small as possible. A facing gauge *G* was inserted into the tank, just clear of the path of the revolving tip. No reading was detected on this gauge, so that the velocity

of the air in the tank could never have exceeded 10 feet per second, corresponding to a reading of $\frac{1}{44}$ inch. Velocities lower than this could not be read on an ordinary facing gauge. The tip speeds used reached 200 feet per second.

TEST OF FACING GAUGE TIPS.

Suppose air to be flowing directly against the mouth of a tube which is connected to a gauge for the purpose of measuring the pressure produced by this velocity. The pressure, h, in feet of air, due to a velocity of V feet per second, is

$$h = \frac{V^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

It is required to find out if the form of the tip affects the correct registering of this amount. If h be expressed in inches of water, as it would be measured in practice, regard must be had to the relative densities of air and water at the time of the experiment.

Let h = pressure measured as a column of air, in feet,
i = the corresponding pressure measured in inches of water,
T = the absolute temperature of air in degrees Fahrenheit,
and h_b = the height of the barometer in inches of mercury.

Then PV being equal to 53.2 T for one pound of air, where P, the pressure of the atmosphere, would be measured by a barometer, the following numerical relation is found between h and i:

$$h = 3.91 \frac{i \times T}{h_b} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Substituting this value of h in equation (1),

$$i = \frac{V^2 h_b}{251.7\ T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This formula gives the reading of the water gauge due to a velocity V for any given atmospheric conditions of barometer and temperature.

For $T = 60°$ F. and $h_b = 30$ inches,

$$i = \frac{V^2}{4{,}370},$$

an approximation near enough for ordinary calculations.

For instance, if the air speed were 200 feet per second, the reading of the water gauge due to this velocity would be—

$$i = \frac{(200)^2}{4{,}370} = 9.15 \text{ inches.}$$

To test a tip, the water gauge connected with the revolving tube is observed, and the reading compared with that calculated as above from the measured velocity of the tip relative to the air in the tank. A correction must, however, be introduced for the vacuum produced by the centrifugal force of the air in the revolving horizontal tube *DE*.

This vacuum, estimated in feet of air column, is

$$h = \frac{v^2}{2g},$$

when v is the speed of the tip. That this is so is mathematically proved in Appendix I. Hence, if the tip measures the pressure due to the air velocity correctly, there should be no reading on the water gauge for any speed (see formula (2))—the vacuum due to centrifugal force just balancing the pressure due to the velocity of the moving tip against the air. The results of the tests of three facing tips are given in Table I, Appendix II.

Experiment No. 597 gives the result of some of the tests made on the standard facing gauge adopted, Fig. 23 in the Plate. The reading of the water gauge connected with the revolving tip is given in column 3, and it will be seen that a slight vacuum was indicated instead of no gauge reading. The calculated vacuum which would be due to an air velocity equal to the tip speed used is given for each case in column 3. The slight vacuum recorded would be accounted for by the assumption that a small stream of air was carried round by the tip. The reading of the water gauge would then be due to the relative velocity of the air and tip, diminished, of course, by the gauge reading due to centrifugal

force; assuming this hypothesis to be correct, the actual air speed in the tank required to produce the vacuum observed, could be calculated, column 5. The maximum velocity does not reach 3 feet per second when the velocity of the revolving tip was 180 feet per second. Hence the standard facing gauge adopted may be taken as correct. Experiments Nos. 602 and 603 were made on the bell-mouthed facing tips, Figs. 24 and 25. The vacuum here recorded in the revolving tube was somewhat greater than in the case of the standard facing gauge for corresponding tip speeds; this, however, is to be expected, since the speed of the air stream carried round with these tips would be greater.

It follows, therefore, that the form of a facing tip, whether conical, parallel or bell-mouthed, for the measurement of velocity, is not of material consequence.

TEST OF SIDE GAUGE TIPS.

From what has been remarked, it will be seen that when a side gauge is rotated, the water gauge reading will be the vacuum due to the centrifugal force of the air in the rotating tube, plus the vacuum produced by the inductive action of the air flowing across the face of the tip. It is the amount of the latter action that is to be determined by experiment.

The results of tests on three forms of tips used as side gauges are given in Table II, Appendix II. Experiments Nos. 604 and 605 were made on the side tips, Figs. 26 and 27. The vacuum recorded in the revolving tube is given in column 3, and in column 4 is shown the calculated vacuum which would be produced in the revolving tube by the action of centrifugal force. The difference between these readings, as given in column 5, is the vacuum produced in the tube by inductive action at each speed. It follows, therefore, as tabulated in column 6, that if these tips be used to measure the pressure in a stream of air flowing through a pipe, by placing them at right angles to the direction of flow, the pressure recorded will be less than the correct amount by a quantity equal to about 45 per cent. of the gauge reading, which would represent the velocity of the air stream in question, column 6. Small differences in the shape of

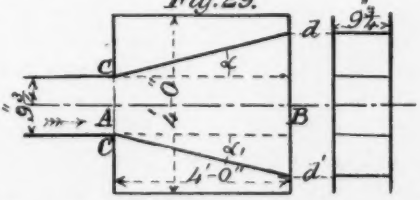
the tips might considerably affect the result. Experiment No. 606 was made on the standard side gauge adopted, shown at Fig. 28. The vacuum recorded in column 5 would be accounted for by an air velocity in the tank itself, of about the same amount as that produced by the bell-mouthed facing gauges (experiments Nos. 602 and 603). It appears reasonable to assign the small reading obtained to that cause, and to assume that the inductive action is completely neutralized by this gauge. A side gauge will indicate correctly even if the direction of the flow of the air makes a considerable angle with the face of the gauge. This can be proved by moving the gauge in the air stream from side to side. For low velocities, such as obtain in mine drifts, the form of gauge is unimportant since there is practically no indication on the gauge due to velocity, or any inductive action to be guarded against; but in the case of blast or cupola fans the conditions are very different, and a correct side gauge is essential.

EFFICIENCY OR EXPANDING CHIMNEYS.

The authors made a series of tests, which are not, however, to be regarded as exhaustive, to obtain the efficiency of an expanding chimney for various angles formed by the sides of the chimney. The kind of apparatus used is shown in Fig. 29.

Air is passed from a fan into a delivery tube $9\frac{3}{4}$ inches square. The end of the tube terminates in two hinged flaps Cd and $C^1 d^1$, which can be set at any desired angle to the center line of the

Fig. 29.



tube. The flaps are also $9\frac{3}{4}$ inches wide and move between two side plates 4 feet long. The side plates can be set up to form an air tight joint. A facing gauge was introduced at A to measure the velocity of the air, and, in addition, a side gauge was applied to measure the vacuum existing at that point due to the action of

the chimney. The vacuum produced was uniform over the whole of the section at A, but the velocity of the air varied considerably. The area of the section was divided into six equal parts, and a velocity gauge was read in each part.

The velocity at the outlet is obtained from a comparison of the inlet and outlet areas;

Let v_i, denote the air velocity at the inlet A,
and v_o the air velocity at the outlet B.

The pressure at the outlet is atmospheric, and the vacuum at A which balances the loss of the head of the air due to velocity, as it passes from the inlet to the outlet of the chimney, is evidently

$$\frac{V_i^2 - V_o^2}{2g} \text{ feet head of air.}$$

Comparing the calculated with the measured vacuum, the efficiency of the chimney, for any particular angle, is obtained. A selection of the results is given in Appendix III.

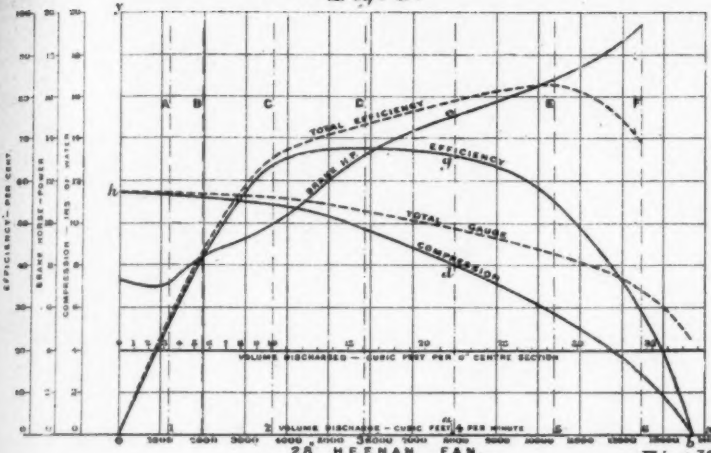
When the chimney is moderately close to the fan blade, the air may not fill the chimney uniformly but flow closer to one side, leaving a partial vacuum on the other side, so that there is an inrush of air at that place. This is particularly the case with chimneys of wide angle when the velocity of the air at the outlet falls below 8 or 10 feet per second. Experiments 58 to 64 were made with the side plates adjusted so that the air escaped uniformly over the outlet area, and an increase of efficiency was the result. A good speed for the air at the chimney outlet is 20 feet per second. In this case, the loss on the water gauge as velocity in air, is

$$\frac{(20)^2}{4,375} = \text{about } \tfrac{1}{10} \text{ inch.}$$

It will be seen from the Table that the efficiency with this speed at the outlet varies between 0.43 for 6 degrees opening, and 0.42 for 15 degrees.

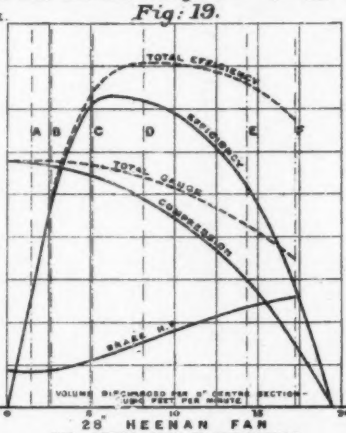
The conclusion to be derived from the tests of expanding chimneys is that in designing a fan chimney, the angle on each side may be as much as 15 degrees without loss of efficiency, but means may have to be taken to give a uniform discharge over the outlet area.

Fig: 3.



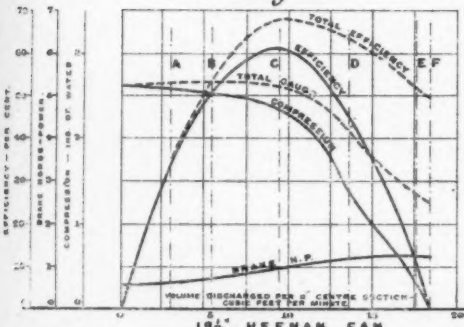
28 HEENAN FAN.
TIP-SPEED 12000 FT PER MINUTE.

Fig: 19.



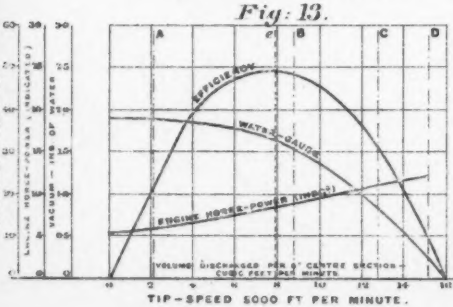
28" HEENAN FAN
TIP-SPEED 6000 FT PER MINUTE

Fig: 18.



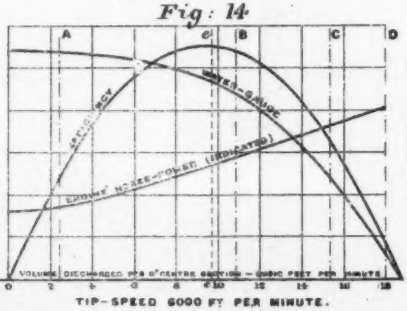
18½" HEENAN FAN.
TIP-SPEED 6000 FT PER MINUTE.

Fig: 13.


EFFICIENCY – PER CENT.
ENGINE HORSE-POWER (INDICATED)
VACUUM – INS OF WATER
EFFICIENCY
WATER GAUGE
ENGINE HORSE-POWER (INDICATED)
VOLUME DISCHARGED PER 0" CENTRE SECTION – CUBIC FEET PER MINUTE

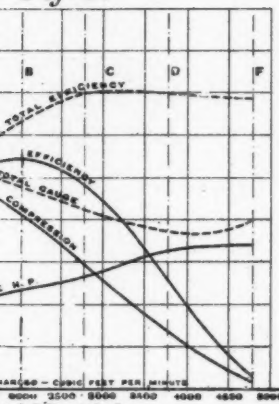
TIP-SPEED 6000 FT PER MINUTE.

Fig: 14



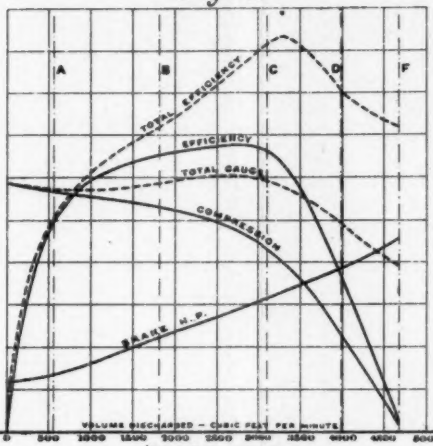
TIP-SPEED 6000 FT PER MINUTE.

Fig: 6.



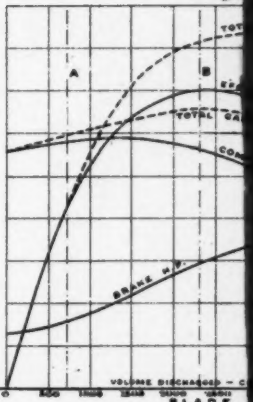
LADE Nº 2.

Fig: 7.



BLADE Nº 3.

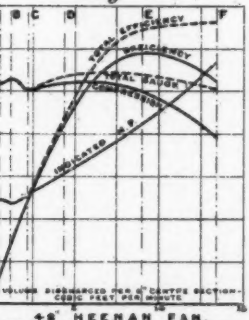
Fig



BLADE

Fig

Fig: 20.



4-6 KEENAN FAN

TIP-SPEED 8000 FT PER MINUTE.

Fig: 10.

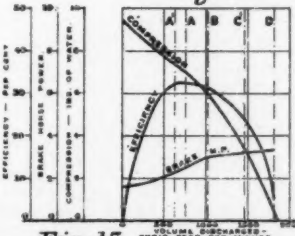
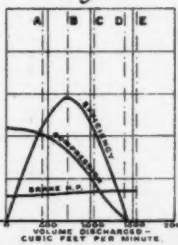


Fig: 11.



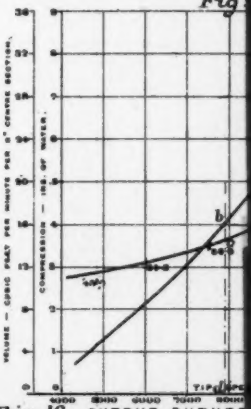
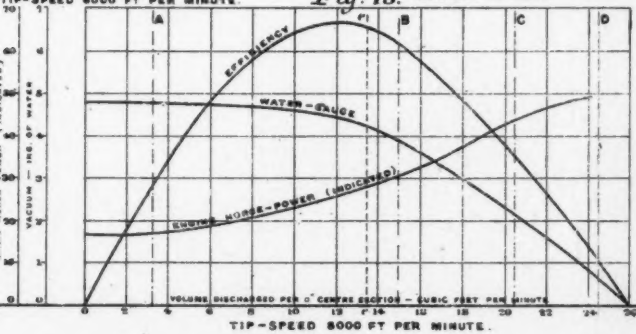
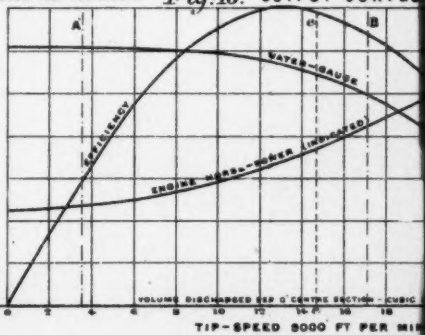


Fig: 15.



TIP-SPEED 8000 FT PER MINUTE.

Fig: 16. OUTPUT CURVES



TIP-SPEED 8000 FT PER MI

PARKEND MINE FAN

CENTRIFUGAL FANS.

Fig: 8.

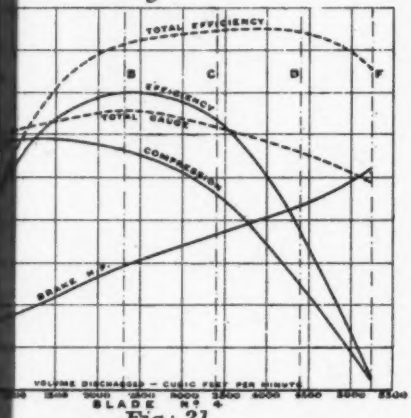
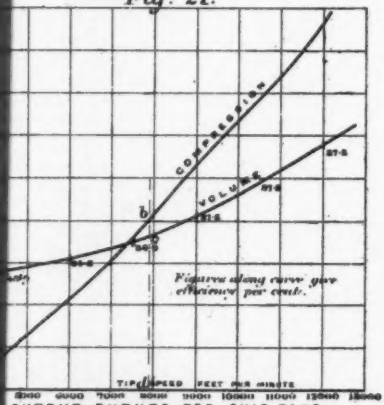
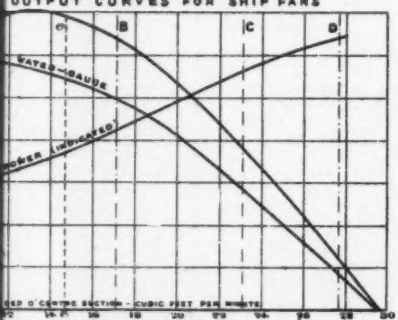


Fig: 21.

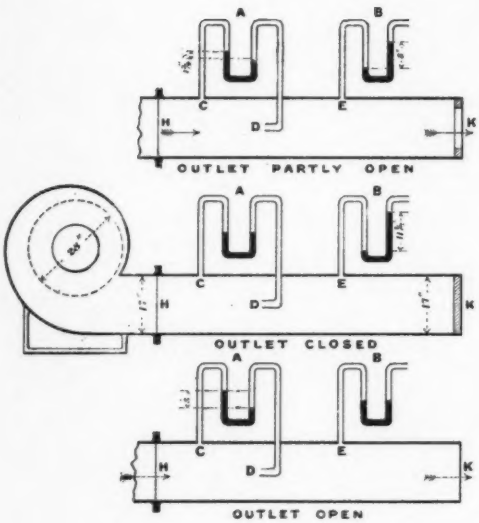


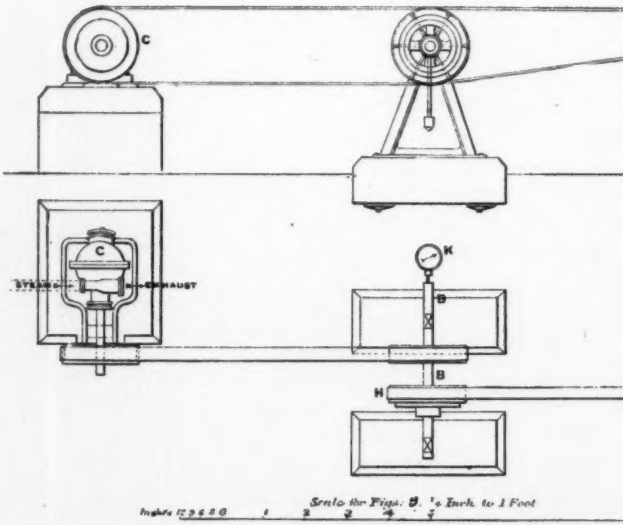
OUTPUT CURVES FOR SHIP FANS



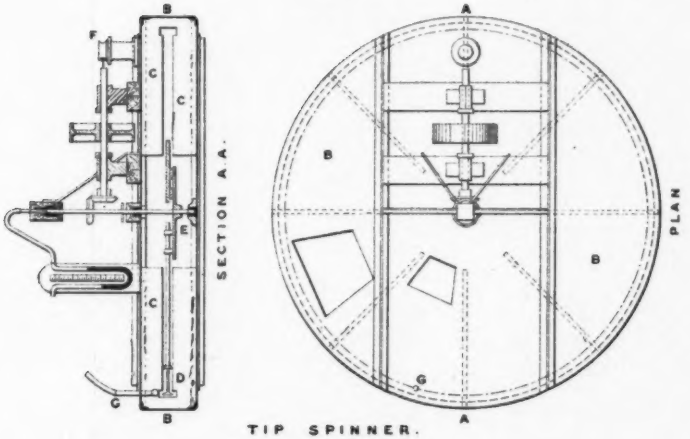
ED 9000 FT PER MINUTE.

Figs. 2.





Figs: 22.



TIP SPINNER.

Scale 3 Feet to one Inch.

Fig: 23.

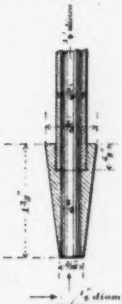


Fig: 26.

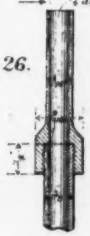
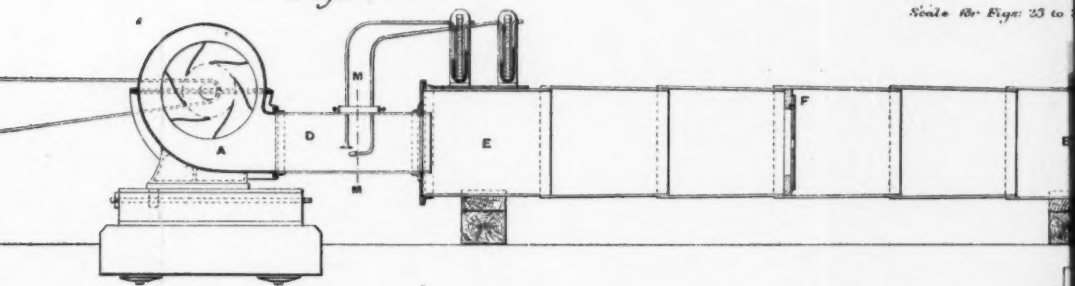


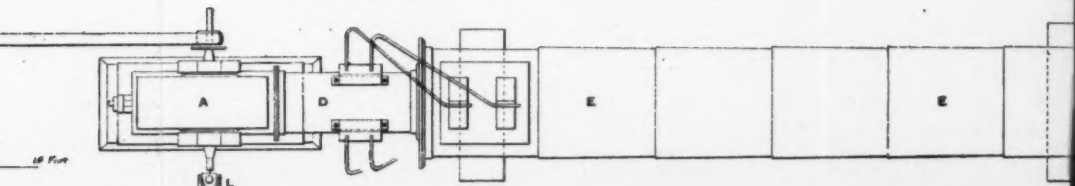
Fig: 27.

Scale for Figs: 23 to

Figs: 9.

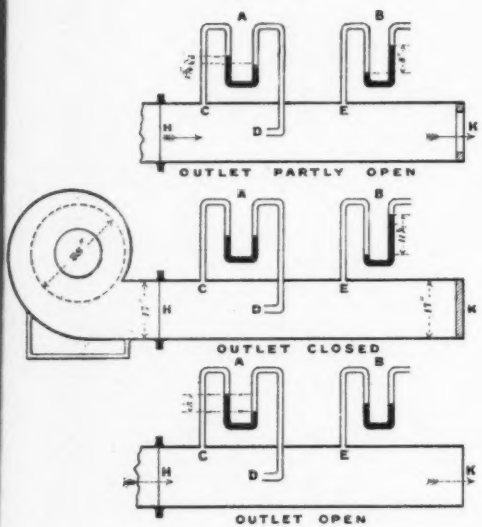


SECTION AT M.M.

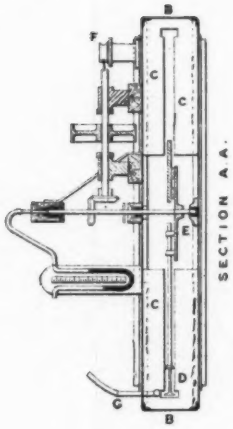


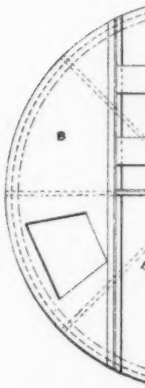
FAN-TESTING APPARATUS.

Figs. 2.



Figs: 22.

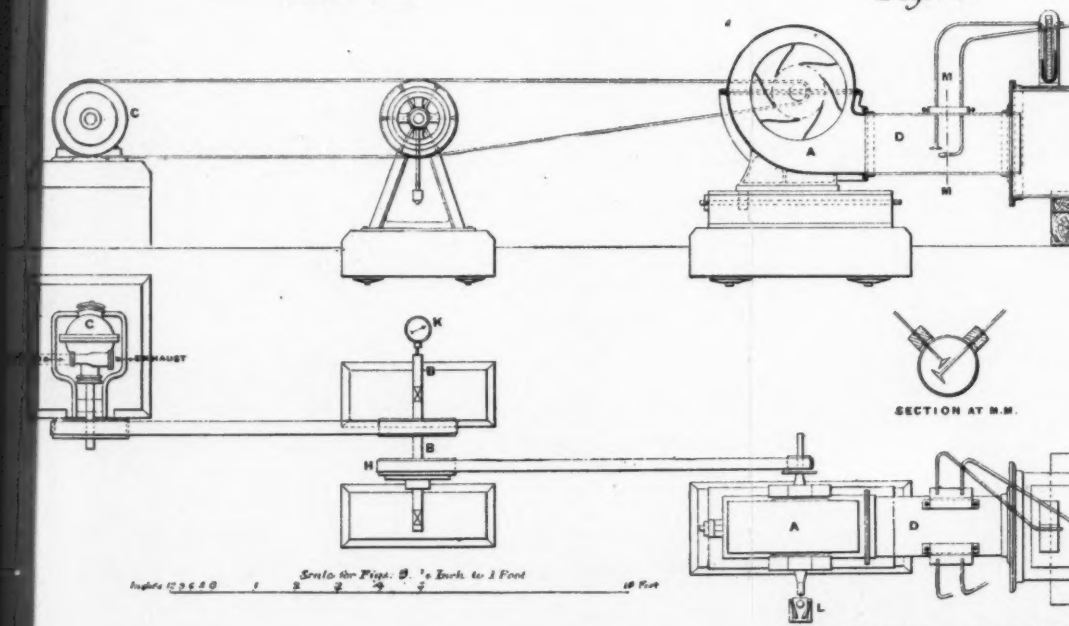




TIP SPINNER

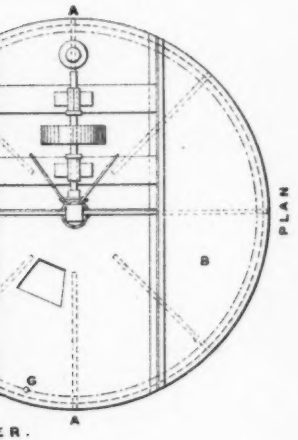
Scale 3 Feet to an In

Figs: 9.



FAN-TESTING APPARATUS

2.



ER.

Inch.

Fig: 23. Fig: 24. Fig: 25.

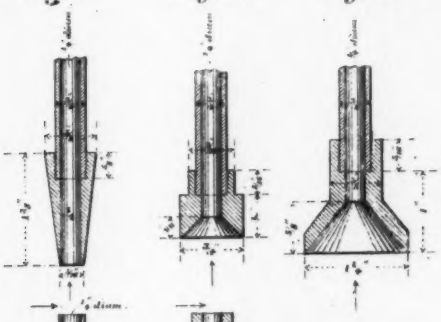
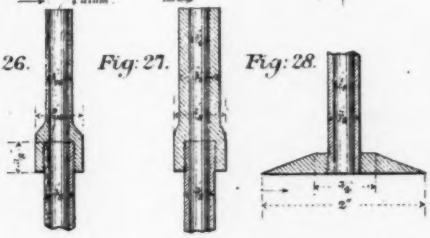
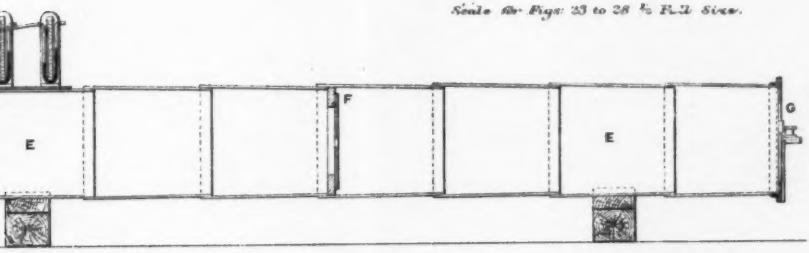
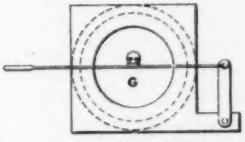


Fig: 26. Fig: 27. Fig: 28.

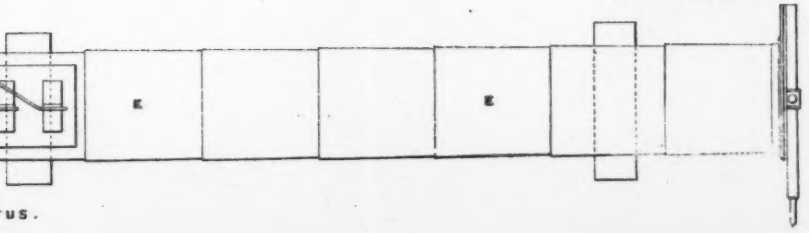


Scale for Figs: 23 to 28 ½ Full Size.





END ELEVATION.

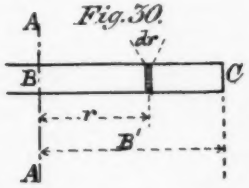


TUS.

Appendix I.

PROOF THAT THE PRESSURE DUE TO THE VELOCITY OF THE TIP IS BALANCED BY THE VACUUM DUE TO CENTRIFUGAL FORCE.

Let BC, Fig. 30, be a tube rotated about the axis AA; it is required to find the pressure on the end of the tube C, due to the centrifugal force of the air enclosed.



Let v be the velocity of the outer extremity of the tube C.

A = the area of the cross section of tube.

G = density of air (weight per cubic foot).

Take a lamina of the air of thickness dr at a radius r.

The volume of this lamina is $A\,dr$, its weight is $G\,A\,dr$, and its velocity is $\frac{r}{R}v$; hence the centrifugal force of the lamina

$$\frac{G\,A\,dr \times \frac{r^2}{R^2}v^2}{g \times r} = \frac{G\,A\,v^2}{g\,R^2}\,r\,dr;$$

hence the centrifugal force of the air in the tube—

$$\frac{G\,A\,v^2}{g\,R^2}\int_0^R r\,dr = \frac{G\,A\,v^2}{2\,g};$$

and the pressure per unit of area at the end C

$$P = \frac{G\,v^2}{2\,g}.$$

Dividing this result by the density, the pressure measured in feet of air column is obtained as under—

$$h = G = \frac{v^2}{2\,g}.$$

APPENDIX II.—RESULTS OF TESTS OF TIPS.

TABLE I.—TESTS OF FACING GAUGE TIPS.

1 No. of experiment.	2 Tip speed.	3 Water gauge reading (vacuum).	4 Gauge reading due to tip speed	5 Calculated air speed in tank.	Remarks.
	ft. per sec.	*inches.*	*inches.*	*ft. per sec.*	
597	79.90	0.01	1.49	0.10	Standard facing gauge, Fig. 23, Plate I. Barometer = 30.17 inches. Temperature = 53° F. $i = \frac{v^2}{4,285}$
	97.00	0.03	2.20	0.65	
	114.00	0.07	3.03	1.40	
	142.00	0.09	4.70	1.50	
	172.00	0.21	6.90	2.70	
602	62.66	0.02	0.89	0.66	Small bell-mouth facing tip, Fig. 24, Plate I. Barometer = 29.38 inches. Temperature = 54.4° F. $i = \frac{v^2}{4,415}$
	103.44	0.15	2.42	3.24	
	132.22	0.26	3.96	4.52	
	169.50	0.42	6.51	5.60	
	180.50	0.48	7.39	5.90	
603	67.16	0.05	1.02	1.56	Large bell-mouth facing tip, Fig. 25, Plate I. Barometer = 29.37 inches. Temperature = 57° F. $i = \frac{v^2}{4,440}$
	99.55	0.11	2.24	2.45	
	132.83	0.23	3.97	4.03	
	164.16	0.35	6.07	4.86	
	184.00	0.43	7.62	5.40	

TABLE II.—TEST OF SIDE GAUGE TIPS.

1 No. of experiment.	2 Tip speed.	3 Water gauge reading (vacuum).	4 Calculated vacuum due to tip speed.	5 Vacuum due to induction.	6 Gauge reading due to induction.	Remarks.
	ft. per sec.				*per cent.*	
604	68.8	1.59	1.07	0.52	49.0	Thin side tip, Fig. 26, Plate I. Temperature = 53.2° F. Barometer = 29.47 $i = \frac{v^2}{4,395}$
	109.5	4.03	2.73	1.30	48.0	
	137.2	6.18	4.27	1.91	45.0	
	160.0	8.50	5.82	2.68	46.0	
	190.4	11.66	8.24	3.42	42.0	
605	81.8	2.24	1.51	0.73	48.0	Medium side tip, Fig. 27, Plate I. Temperature = 57° F. Barometer = 29 63. $i = \frac{v^2}{4,400}$
	99.8	3.20	2.26	0.94	42.0	
	113.2	4.22	2.90	1.32	46.0	
	136.8	6.40	4.25	2.15	51.0	
	167.8	9.65	6.38	3.27	51.0	
606	69.3	1.13	1.09	0.04	3.7	Standard side gauge, Fig. 28, Plate I. Temperature = 44° F. Barometer = 28 89. $i = \frac{v^2}{4,400}$
	81.0	1.58	1.49	0.09	6.0	
	104.0	2.58	2.46	0.12	4.9	
	127.3	3.90	3.68	0.22	6.0	
	151.0	5.57	5.19	0.38	7.3	
	187.0	8.48	7.95	0.53	6.7	

In experiments 604 and 605 a slight correction is necessary in the last column, owing to the air speed in the tank.

Appendix III.

EFFICIENCY OF AN ÉVASÉE CHIMNEY FOR VARIOUS ANGLES.

No. of experiment.	Angle of sides of chimney.	Velocity of the air.		Observed vacuum in inches at A.	Calculated vacuum in inches at A.	$k = \frac{\text{obs. vac.}}{\text{cal. vac.}}$
		At inlet A.	At outlet B.			
10	3°	32.1	21.2	0.02	0.13	0.15
12		40.8	26.9	0.11	0.21	0.51
14		65.7	43.4	0.33	0.55	0.60
16		79.8	52.6	0.55	0.82	0.67
18	6°	32.0	15.7	0.07	0.18	0.39
20		49.2	26.1	0.20	0.42	0.48
22		67.9	33.3	0.45	0.80	0.56
24		79.0	38.7	0.67	1.07	0.57
26	9°	33.9	13.2	0.05	0.22	0.22
28		50.7	19.8	0.18	0.49	0.36
30		69.0	27.0	0.40	0.91	0.44
32		82.1	32.1	0.63	1.30	0.48
34	12°	32.3	10.4	0.07	0.21	0.33
36		50.3	16.3	0.18	0.52	0.35
38		65.2	21.1	0.39	0.96	0.41
40		80.0	25.9	0.60	1.30	0.46
42	15°	33.2	9.1	0.11	0.23	0.47
44		48.6	13.3	0.15	0.50	0.30
46		67.4	18.5	0.40	0.90	0.42
48		77.0	21.2	0.52	1.25	0.42
50	18°	31.6	7.5	0.06	0.31	0.28
52		48.1	11.4	0.18	0.50	0.36
54		63.8	15.2	0.35	0.88	0.40
56		79.0	18.8	0.49	1.34	0.37
58	6°	33.1	9.7	0.10	0.23	0.44
60		48.0	14.1	0.22	0.48	0.45
62	21°	67.3	19.8	0.54	0.94	0.57
64		81.2	23.8	0.75	1.49	0.52

ON CRANK SHAFTS.

By John H. Macalpine, Esq., Associate Member.

[Continued from page 83, Volume IX.]

STRESSES DUE TO STEAM AND INERTIA.

After the earlier discussion of the effect of very bad distribution of power among the three cylinders, the subject may be dismissed very shortly here.

The only two positions of the L.P. crank which we have to consider are when it is about 27° before top center and when it is about 27° past bottom center. When the shaft is in line and for equal distribution of power, the former position gives maximum stresses for all parts of the L.P. crank shaft except the forward main journal, for which the stress is a maximum in the latter position.

L.P. Crank About 27° Before Top Center.

In this position the turning moments due both to steam and inertia are positive at all parts of the L.P. shaft—excepting the slight negative turning moment due to the inertia of the H.P. and I.P. engines for the 2d order of cranks, Fig. XI, with which we need not deal. The two turning moments thus augment one another; the bending moments are also of one sign. Thus any increase or diminution of the moments due to steam will produce an increase or diminution in the total stresses, but the proportionate change will be diminished—for the larger values of $\mathfrak{M}$, much diminished—by the fact that steam only produces a part of the stress. For an equal distribution of power the H.P. and I.P. cylinders each give a larger steam turning moment than that from the L.P.; so that when the increase of power is in the L.P., not only is the sum of the turning moments from the two forward cylinders decreased, but the total turning moment from all three cylinders is diminished—however the sacrifice is

made by which the L.P. power gains. Fig. VII shows that the reaction at bearing A from steam is not large at ordinate 37, but that the increase of power in the L.P. will increase it. From Tables XIII and XIV we see that, for the parts of the shaft there considered, the steam bending moment forms much less than half of the total bending moment—except for very small values of $\mathfrak{M}$*. This is also true for the L.P. forward web. Only in these cases where the arms for the bending moments are large is a 50 per cent. increase of power in the L.P. cylinder able to more than counteract the diminution of turning moment and produce a slight increase of stress.

If the L.P. power remains unchanged and redistribution of power is entirely between the H.P. and I.P. cylinders, increase of stress will in some cases result. The worst case will be for the 1st order of cranks and augmented power in the I.P. at the expense of the H.P. The turning moments will be augmented most, consequently the greatest rise of stress will be for small values of $\mathfrak{M}$ and F_m; that is, when the inertia moments are small and the bending moments relatively unimportant. Taking the extreme case of $\mathfrak{M} = 2$ and $F_m = 10$, a 50 per cent. increase of the I.P. power, and corresponding diminution of that from the H.P., would cause a total increase of stress in the crank pin of only 9 or 10 per cent. This percentage will rapidly diminish as $\mathfrak{M}$ and F_m increase.

Diminution of power in the L.P. cylinder, as in assumption (c), Table VI, was rejected in the earlier discussion for a reason which is equally valid here. If it did occur, it is evident that the increased turning moment and diminished bending moment would have most effect where the arms for the bending moments are small. The greatest rise is somewhat larger than for the L.P. power constant, but is never important.

L.P. Crank 27° Past Bottom Center.

For this position we need only consider the effect on the stresses of the forward L.P. main journal. Here we will find that,

* $G/F_m = .2044\ \mathfrak{M}$.

for the bearings in line and equal distribution of power, the stress never rises much above 13,000 pounds per square inch. The greatest stress is for the 1st order of cranks.

The turning moments from steam and inertia are both positive; but the steam reactions on the L.P. pin and after bearing A oppose those from inertia. Thus for the larger values of $\mathfrak{M}$, when the inertia reactions are greater than those from steam, increase of power in the L.P. will decrease those reactions, and vice versa. For the smaller values of $\mathfrak{M}$ the opposite effects are produced. Thus the total effect of increase of power in the L.P. would be to decrease the sum of the turning moments from the H.P. and I.P. and, for the larger values of $\mathfrak{M}$, decrease the bending moments: therefore the maximum stress will fall. For the smaller values of $\mathfrak{M}$, the bending moments in the forward journal produce only a small effect on the stress; and in no case will an increase in the augmented equivalent turning moment result.

If, for the 1st order of cranks, the H.P. power is increased 50 per cent. and the I.P. diminished correspondingly—the L.P. remaining the same—the stress will rise to about the 15,000 pounds per square inch allowed above when $\mathfrak{M} = 2$ and $F_m = 10$. But as we increase F_m and $\mathfrak{M}$, the increase of stress diminishes and the total falls farther and farther below 15,000 pounds per square inch.

Decrease of power in the L.P. would at worst give about the same increase of the steam turning moment as in the case just considered. But the bending moment from steam would be largely diminished; producing, as before, an effect on the total bending moment depending on the value of $\mathfrak{M}$. It is still only in the case of the smaller values of $\mathfrak{M}$ and F_m that the total stress does not fall decidedly under 15,000 pounds per square inch.

If in this discussion we had supposed bearing B removed, more important results would not have been reached. The statement can readily be verified, and there is little object in extending the discussion farther than to remark, that want of alignment causes the most considerable rise of stress for the larger values of $\mathfrak{M}$ and F_m; and, for these same values—as before noted—a change in the distribution of power can cause little effect, since the steam moments are then relatively less important.

There is then, I think, no sufficient reason for making any change in the curves, Figs. XIV to XXI, for other distributions of power than that supposed at first.

The question of want of alignment of the bearings has been treated in a similar manner to that explained in the earlier part of the paper, and the same general considerations apply here.

Table XIX exhibits the effect of bearing B being out of line by $\frac{1}{1000}$ of an inch, when we consider a similar engine of one inch stroke. The table is constructed and is to be used in the same manner as Table VII: thus the effect of any derangement of bearings may readily be estimated. It has been considered sufficient to calculate the various values for $\mathfrak{M} = 5$, 8 and 10.

Lines (a_1), (a_2) and (a_3)—referred to afterward as lines (a), the other lines being similarly designated—give, as before, the actual increase of reaction at bearing A in pounds per square inch, supposing the pressure uniformly distributed. The influence of the webs is also allowed for.

Lines (b), (b′), (c) and (c′). These figures are for the 1st order of cranks and the positions of L.P. crank given in Tables XIII and XIV. In lines (c′) the multiplier for change of section is cancelled.

Lines (d) and (d′). These figures are for the 1st order of cranks and the positions of the L.P. crank given in Tables XIII and XIV. Here increase of reaction, if vertical, will diminish the turning moment in the crank pin. Especially where b is small will the direction of the reaction at bearing A affect the value of the stress in the pin. To exhibit this effect two lines of figures are given. In both, the part of the reaction induced when the bearings are in line is supposed to remain in the same direction as that taken in calculating the curves of shaft dimensions; that is, parallel to the L.P. connecting rod. The reaction due to want of alignment is supposed, in the upper line of figures, to be vertical; in the lower line it is taken parallel to the crank, so that it produces no diminution of turning moment, and thus adds more quickly to the increase of stress. The upper line represents more nearly what will usually occur, as want of alignment is most

frequently produced by wear, in which case the direction of the induced reaction will not as a rule be far from the vertical. Accidental causes, such as the grounding of the ship, especially if the engine is of a very light type, might readily produce such effects as are given by the lower line of figures.

Examination of the stress, with the shaft in line, when the L.P. crank is 27° past bottom center, near which a maximum always occurs, shows that the worst case is for the 1st order of cranks. Here the crank pin turning moment is greater than when the L.P. crank is about 27° before top center. But as the steam and inertia forces on the L.P. crank pin now oppose one another, the bending moment is much smaller even when $\mathfrak{M} = 10$—for which value the inertia force is much the larger of the two. For $\mathfrak{M} =$ about 4, the bending moment vanishes on account of the equality of the opposing reactions from steam and inertia. The total stress in all cases falls considerably below 15,000 pounds per square inch.

The rate of increase of stress per unit increase of reaction at bearing A, when that reaction is supposed vertical, is never much greater here than for the case represented by the upper lines (d) except when F_m is near its lowest values. Even when $F_m = 100$ and $\mathfrak{M} = 10$ the rate of increase in upper lines (d) is a little smaller than in the case now spoken of; but as $\mathfrak{M}$ diminishes, this condition is reversed on account of the more rapid fall of bending moment now produced. The rate of increase in the present case will always be somewhat less than for the conditions supposed in the lower lines (d), except where F_m is small and $\mathfrak{M}$ large. Of course the reaction due to want of alignment may be inclined so as to give a greater rate of increase than the vertical reaction would, when the crank is about 27° past bottom center; but, considering that the stress is now considerably lower than 15,000 pounds per square inch for the shaft in line, we may safely look on the cases given as the worst. The questions here raised will be reverted to shortly, in a note on page 323, as they are of some importance.

Lines (e) and (e′). These figures are for the 1st order of cranks and for the L.P. crank above 27° before top center. The stress is measured near the crank pin end of the web, as

there the turning moment is the largest; but the reaction due to want of alignment is supposed to be along the web, so that the turning moment is not affected by it. This corresponds with the lower lines (d), and if two lines had been given the same kind of difference would have showed itself. There is no multiplier for change of section.

If we consider the case of the L.P. crank about 27° past bottom center, we have the stress highest for the 1st order of cranks when the bearings are in line. It is less than the figures given in lines (e′) by from 1,400 to 2,400 pounds per square inch when $F_m = 100$, depending on whether $\mathfrak{M} = 10$, 8 or 5. For $F_m = 10$ and $\mathfrak{M} = 10$, the stresses are practically the same in both cases. When the reaction at bearing A is increased, both bending and turning moments now rise; and if we suppose the direction of this added reaction to be at the angle which will make the rate of increase of total stress a maximum, the rate here is higher—for all but the smallest values of F_m, only a little higher—than for the case in lines (e). Hence, that in the table is given as the worst case.

Lines (f) and (f′). These figures are for the 1st order of cranks and the L.P. about 27° past bottom center. The reason for the stress being highest at this point is the large value of the inertia turning moment. As the ratio of turning to bending moment is larger here than when the L.P. crank is about 27° before top center, it can be shown to follow that the rate of increase of stress per unit increase of reaction at bearing A is larger in the latter case. But, as the stress for the shaft in line is then much smaller than that given in lines (f′), the case tabulated is the worst.

NOTE.—The following will make the meaning of some of the statements made in the last few paragraphs clearer.—

For the round parts of the shaft we have

$$T_a = B + \sqrt{\left[B^2 + \left(\frac{10}{7}T\right)^2\right]}, \text{ and}$$

$$\frac{dT_a}{dB} = 1 + \frac{1}{\sqrt{\left[1 + \left(\frac{10}{7}\frac{T}{B}\right)^2\right]}} + \frac{\frac{10}{7}\frac{T}{B}\cdot\frac{dT}{dB}}{\sqrt{\left[1 + \left(\frac{10}{7}\frac{T}{B}\right)^2\right]}}.$$

For the parts lying along the center line of the shaft, $dT/dB = 0$. The first two terms on the right-hand side of the equation then show that the value of dT_a/dB depends on the ratio T/B; that is (see equation (4) above) on the directions of the principal planes of stress. As the ratio T/B is larger, and consequently the principal planes of stress more nearly at 45° to the axis of the shaft, dT_a/dB is smaller. Hence the statement under lines (f) and (f′).

The extreme cases are :—

1st. $T = 0$. Here $dT_a/dB = 2$, and the principal plane of stress normal to the shaft axis is the only one of the two across which there is any normal traction. As no turning moment is induced, the principal planes of stress retain their position and the ratio of the increase of T_a to that of B remains constant for finite changes of these quantities—as is otherwise well known.

2d. $B = 0$. $dT_a/dB = 1$, and the newly induced stress does not add directly to that already existing since the principal planes of stress are now inclined at 45° to the axis of the shaft. This ratio gradually increases for finite changes of stress as the bending moment induced affects it more and more; or, in other words, one of the principal planes of stress is gradually turned toward the shaft axis.

This presents, in an instructive light, some statements which have been made earlier.

We see, then, that for the main journals the greatest rate of increase of stress for want of alignment can never occur at the points of the revolution at which we have the stress a maximum, but must always be at those points where there is no turning moment.

For the crank pin we have $dB/df = b/2$, very nearly, if the angle between the added reaction and that with the shaft in line is not large. All multipliers are neglected as this is only an illustration. As before, f is the reaction at bearing A.

Also $dT/df = \mp .5 \sin 27°$ for one inch stroke, if the crank is at 27° before top center and df acts vertically—the upper or lower sign being used according as the increase of reaction diminishes or increases the turning moment. Thus, we get from the above equation,

$$\frac{dT_a}{df} = \frac{b}{2} + \frac{b}{2\sqrt{\left[1 + \left(\frac{10\,T}{7\,B}\right)^2\right]}} \mp \frac{.324\,\frac{T}{B}}{\sqrt{\left[1 + \left(\frac{10\,T}{7\,B}\right)^2\right]}}.$$

The first term on the right hand side is the same for each case in the text.

The second term is larger when the L.P. crank is about 27° before top center than when it is 27° past bottom center, because in the latter case the ratio T/B is much larger—especially for the smaller values of $\mathfrak{M}$. This is true whether the reaction is supposed vertical or parallel to the crank. To the degree of approximation of the equation this part of dT_a/dB is not affected by the direction of the reaction; this is, however, not quite true.

The third term should obviously be omitted when the increase of reaction is parallel to the crank. It is only relatively large when b is small.

As we have seen, when the reaction at bearing A is determined by the L.P. steam

forces principally, increase of reaction diminishes the L.P. crank pin turning moment. But in the early part of the upward revolution of the L.P. crank, when ☊ is large, this reaction is principally determined by the inertia force on the L.P. pin; and as this opposes the steam force, increase of reaction will increase the turning moment. It is to be noted that both the steam and inertia turning moments are positive for the L.P. pin at this position of the crank. For small values of ☊ the steam force is the controlling factor, and increase of reaction diminishes the turning moment; then the case of the L.P. crank 27° past bottom center ceases to be of any importance. As used here, "increase of reaction" means increase of its absolute amount.

This will be found to bear out the statements in the text; which, however, were founded on numerical results.

For the webs, unless the bending moments are unimportant relatively to the turning moments, the greatest rate of increase of stress must occur when the L.P. crank is near top or bottom center. It is only there that the bending moments for the shaft in line are very effective in raising the total stress; and the greatest effect of the increased reaction from want of alignment will be due to the change of bending moment rather than of turning moment which it produces, unless the arm for the bending moment is very small. As we have seen, at any one point of the revolution, the rate of increase of stress per unit increase of reaction is constant for this part of the shaft, in each particular case.

For the forward L.P. web, when the crank is 27° before top center and the increased reaction parallel to the crank is Δf, which may be any finite quantity, we have,

$$\text{Increase of stress} = \frac{6\,\Delta f \times K}{\epsilon\, e^2}, \text{ or}$$

$$\text{Rate of increase of stress} = Const. \times K.$$

If the L.P. crank is a little past bottom center, and ☊ so large that increase of reaction increases both turning and bending moments, we have—χ being the angle at which the new part of the reaction is inclined to the web—

$$\text{Rate of increase of stress} = \frac{6\,K\cos\chi}{\epsilon\, e^2} + \frac{6 \times .45 \sin\chi}{\epsilon^2\, e}$$

$$= Const.\ (K\cos\chi + \frac{e}{\epsilon} \times .45 \sin\chi),$$

where "*Const.*" has the same value as above. $.45 \sin\chi$ is the arm of the turning moment, as the stress is measured a little within the radius of the pin.

The latter rate is a maximum when

$$Tan\,\chi = \frac{.45\,e}{\epsilon\, K},$$

and it has then the value

$$Const. \times K \sqrt{\left\{1 + \left(\frac{45\,e}{\epsilon\, K}\right)^2\right\}}.$$

The ratio of the two rates is thus

$$\sqrt{\left\{1+\left(\frac{.45\,e}{e\,K}\right)^2\right\}},$$

which is only sensibly above unity when K is very small, that is, when F_m is small.

$F_m = 10$, $\mathfrak{M} = 2$, $K = .334$, $e'/e = 1.4$, and the above ratio $= 1.39$.

$F_m = 100$, $\mathfrak{M} = 10$, $K = 1.849$, $e'/e = 1.7$, " " " " $= 1.01$.

Perhaps it is well to explain here, more particularly than has been yet done, the exact procedure in calculating the rise of stress in the round parts of the shaft, Tables VII and XIX. The simplest case is that of the aft main journal and it will be sufficient for illustration. The problem as first stated was—

"What want of alignment would increase the pressure on the after main bearing, A, so that the maximum stress on the aft main journal would increase 10 per cent. at the point of revolution for which the calculation of the curves has been made: the previous multiplier for errors of adjustment being retained?"

The retention of these multipliers make the figures in the tables represent the worst we need suppose possible, and considerably worse than is at all likely to occur.

T_1 = the turning moment; which is not affected by want of alignment. $T_1 = 1.216\ F_m$ (from Table I) or $= 1.056\ F_m + .389\ G$ (from Tables XIII and XIV).

$f_1\,k$ = the bending moment for the shaft in line. $f_1 = .55\ F_m$ (from Table I) or $= .325\ F_m + .618\ G$ (from Tables XII and XIV).

f_a = increase of reaction at bearing A due to want of alignment.

To allow for the effect of change of section we may write $1.1\,f_a\,k$ for the total increase of bending moment. Then we have, $B_1 = (f_1 + 1.1\,f_a)\,k$ = total bending moment.

The moment of resistance against torsion of a shaft which has a hole of half the outside diameter bored axially through it, as supposed here, is

$$\frac{\pi}{16}\times\frac{(a^4 - .5^4\,a^4)}{a}\,U = .184\,U a^3,$$

where U, as before, is the maximum shearing stress induced, and a the outside diameter of the shaft. We may write this λa^3, where $\lambda = .184 \times 11{,}500 = 2{,}116$ for steam stresses alone, and $= .184 \times 15{,}000 = 2{,}760$ for steam and inertia stresses. As the stress is now to be raised 10 per cent. we have

$$\sqrt{(B_1^2 + T_1^2)} + B_1 = 1.1\,\lambda a^3.$$

Transposing and squaring

$$B_1^2 - 2.2\,B_1\,\lambda a^3 + 1.21\,\lambda^2 a^6 = B_1^2 + T_1^2,$$

and $$2.2\,B_1\,\lambda a^3 = 1.21\,\lambda^2 a^6 - T_1^2.$$

Or, $$2.2\,(f_1 + 1.1\,f_a)\,k = 1.21\,\lambda a^3 - T_1^2/\lambda a^3.$$ Similarly, for the shaft in line $$2\,f_1\,k = \lambda a^3 - T_1^2/\lambda a^3.$$

By subtraction and transposition

$$2.42\,f_a\,k = .21\,\lambda a^3 - .2\,f_1\,k, \text{ or } f_a = \frac{.21\,\lambda a^3 - .2\,f_1\,k}{2.42\,k}.$$

If in expression (15), we suppose $l = n = p = 0$, we get

$$f_a = \frac{5.301 \times 10^6 \, a^4 m}{b^3}.$$

Equating those two expressions for f_a, we get

$$m = \frac{1.559 \times 10^{-8} b^3}{a^4 k} (1.05 \lambda a^3 - f_1 k).$$

From this, in each case, the value of m which will produce a 10 per cent. rise of stress can readily be calculated, and hence the mean rate of rise of stress per .001 error of bearing B. Before entering the results in Tables VII and XIX they must be divided by the factors for the influence of the webs, as fully explained above

I will now transform the figures in Table XIX in a similar manner to that used in calculating Table VIII.

If we estimate the pressure on bearing B, which may be done by means of the equation

$$f_B = \frac{1}{40} (6 F_1 - 23 F_2 - 29 F_3),$$

where f_B is the total reaction on this bearing, we will find very approximately the following ratios for the maximum reaction on bearing B, to that on bearing A given by the value in Table XV, line (m), columns III and IV.

Position of L.P. crank.	About 27° before top center.		
𝔐	10	8	5
Ratio. 1st order of cranks	1.003	.916	.700
Ratio. 2d order of cranks	1.437	1.544	1.814

These ratios, it will be noticed, do not depend on F_m but on the ratio G/F_m or 𝔐 only.

Except for the very highest value of 𝔐, bearing A has a higher total pressure than bearing B for the 1st order of cranks. Where the latter, whether divided into two or not, has—as is usual—a good deal greater surface than bearing A, the pressure per square inch will be much smaller. Thus the limit of want of alignment which could occur before a bearing would heat is determined here by the behavior of bearing A; even though, as on page 710,

Vol. VIII, we may suppose the intensity of pressure at B to rise 1.5 times as quickly as at bearing A.

With the 2d order of cranks, if bearing B has one and one-half times the area of bearing A, the pressure will vary from a little under 400 pounds per square inch to somewhat higher, depending on the value of $\mathfrak{M}$. This bearing can safely be allowed to run at a rather higher pressure than bearing A; for, as noted on page 63, Vol. IX, an intermediate bearing gets its highest pressure only when the angular displacement of its journal is small and thus better disposed to retain the lubricant. But it should be watched, especially when the cranks are in the 2d order for the engine running ahead, that bearing B has never an excessive pressure. It is bearing B which now sets the limit to the possible want of alignment which could occur before a bearing would heat, and this will require a much smaller rise of reaction at bearing A—that is, a smaller rise of the bending moments stressing the L.P. shaft—than in the case of the 1st order of cranks. The 1st order, then, endangers the life of the shaft most in this respect, and therefore Table XX, deduced from the figures in Table XIX, is decidedly the worst case. It will also be remembered that in taking this order of cranks we start from the highest stresses for the shaft in line.

The intensity of pressure on the bearings allowed in Table XVII being higher than that in Table V, if we used the same supposition as to the limiting pressure which could be attained without heating as was done on p. 710, Vol. VIII (that is, a maximum pressure of 600 pounds per square inch), the direction of pressure would still be alternating. Nevertheless, considering the greater flexibility of the shafts given in Figs. XIV to XXI, and the consequently smaller area of high pressure in the bearings, it may be doubted if the average pressure could be raised further in their case. But to set a limit which, it can hardly be doubted, is at least as bad as that supposed in Table VIII, it has been assumed in calculating Table XX that the maximum pressure runs up to 700 pounds per square inch. To give this intensity of pressure at bearing A requires a rise of 300 pounds per square inch.

In Table XIX for $F_m = 10$ and $\mathfrak{M} = 5$, .001 inch error of bear-

ing B will cause a rise of pressure of 509 pounds per square inch at bearing A; hence 300 pounds per square inch will be produced by an error of $300 / 509 = .59$ of one thousandth of an inch, and the stress induced in the forward L.P. web will be

$$12{,}035 + .59 \times 9{,}147 = 17{,}432.$$

The other values in Table XX were similarly calculated.

It should be noted that the case of the L.P. forward journal is a little different from that just stated and given in Table XX. The maximum pressure on the upper brass of bearing A, when steam and inertia stresses are considered, is greater than the maximum pressure on the lower brass; and it is with the former alone we have dealt. The maximum stress takes place in the forward journal when the L.P. crank is about 27° past bottom center; and, if $\mathfrak{M}$ is higher than about 3.5, the pressure at bearing A is on the lower brass. Raising bearing B to increase the maximum reaction (on the upper brass) as supposed, will decrease reaction on the lower brass and consequently decrease the maximum stress in the forward journal. Thus, to be correct, we should either count the rise of stress in this journal from the considerably lower maximum which occurs when the L.P. crank is approaching top center, or the rise of pressure in bearing A from the somewhat lower maximum exerted on the under brass. But as the conditions for seizing are so difficult to ascertain, it is not worth while to enter into refinements. The true evaluation would evidently give a greater deflection at bearing B for the limiting condition than is given in Table XX.

The two conclusions to be drawn from Table XX are:—

1st. For small values of F_m and $\mathfrak{M}$ the rise of stress under the conditions supposed is not very excessive, and would not damage a sound shaft.

2d. The elasticity of the shafts is so great that, except for very short stroke engines or engines with low values of F_m and $\mathfrak{M}$, the bearings could not be supposed to get out of line by the required amount; and consequently the shafts are strongly protected against such large rise of stress as is given by the table.

It seems hardly necessary to repeat an analysis such as was given of some of the results in Table VIII. If made in any particular case with the deflection at bearing B reduced to correspond to $\frac{1}{40}$ or $\frac{1}{50}$ inch for the actual stroke of engine—more than should ever occur in practice—this analysis will, I think, fully bear out the two above conclusions. The deflections even for $\mathfrak{M} = 5$ are seen to be greater than those in Table VIII, and they rapidly rise as $\mathfrak{M}$ increases.

While we may perhaps conclude that with a perfectly sound shaft of a good quality of material the stresses are not likely to rise higher than it can safely stand, any particular shaft may be imperfect notwithstanding the care exercised; and it will sometimes be impracticable to prevent the occurrence of sensible errors of alignment, especially in times of war or other hard service, when a breakdown might be calamitous. Thus it will always be well to examine what stresses may occur, and this can readily be done by means of Tables XIX and XX.

As the shafts in Figs. XIV to XXI are deduced after assuming particular constants, the truth of Tables XIX and XX is limited by these assumptions: thus, it is important to examine what changes will be made in the various lines of the table for a change of shaft diameter.

If we change to a shaft of higher stress, a is diminished; but b is very little reduced, as the pressure on the bearings can not be run up indefinitely. The turning moments are unaffected. The bending moments in Tables XIII and XIV change only so far as b does, for the arms of the bending moments stressing each part of the shaft are nearly proportional to b. Thus in the round parts of the shaft, for the bearings in line, the stress is inversely as a^3 very nearly; and the positions of the principal planes of stress are little changed. From the last statement it follows that rise of stress due to want of alignment may be taken as proportional to rise of bending moment and inversely proportional to a^3. A similar conclusion holds with regard to the webs.

The reaction at A for a given derangement of the bearings depends, neglecting any change in the factor for the influence of the webs, on the ratio a^4/b^3; and, consequently, the bending moments

induced, on a^4/b^2. Thus, the change in rise of stress from want of alignment will be nearly proportional to a/b^2. This ratio will not usually change much; so that, while the stresses for the shaft in line vary rapidly as a is changed, those due to want of alignment remain nearly constant.

Do not these facts throw considerable light on the surprising immunity from breakage which seems to be enjoyed by the crank shafts of torpedo boats? I say "seems to be enjoyed", for I am not aware that any statistics to show the real facts have ever been collected; but I have often been surprised at the rarity of such an occurrence considering the very high stress under which these shafts work. No doubt every precaution is used to obtain the best material and the most careful manufacture, and the engines are not exposed to the same continuous hard work as those in the merchant marine. They have a high value of $\mathfrak{M}$ and also, as a rule, a comparatively high value of F_m. So that, if the shafts were determined by Figs. XIV to XXI, the error of bearing B—when the conditions were such as supposed in Table XX—would be comparatively large, notwithstanding their short stroke.

But with the smaller shaft which would actually be fitted, the stresses due to want of alignment will not increase; that is, the relative importance of this part of the stress will diminish with the ratio a^4/b^2. To take a numerical example.—If the stress for the shaft in line is raised from 15,000 to 20,000 pounds per square inch while b is not altered, a would have to be diminished in the proportion $(15/20)^{\frac{1}{4}}$ and the relative importance of the bending moments due to want of alignment would fall in the proportion of $(15/20)^{\frac{1}{4}} = .68$.

It must not be supposed, however, that the diminution of crank shaft diameters in ordinary practice is here advocated as safe. It is well known that under the conditions as to manufacture and use holding in the merchant marine, the diameters of crank shafts cannot be diminished much below that given by the various rules issued by the Boards controlling merchant practice, without greatly endangering the life of the shaft. All that is proved is that, where the diameter of the shaft is diminished, the maxi-

mum stress arising from steam, inertia, and want of alignment does not rise so fast as $1/a^3$.

Following the earlier procedure, let us now examine the stresses induced when bearing B is removed.

Fig. XXIII gives the turning moments on the L.P. crank pin due to steam. The dot-and-dash line is the sum of the H.P. and I.P. turning moments, copied from the dot-and-dash line of Fig. III. It was thought well to add the crank pin turning moment for the bearings in line, which was only shown approximately by the dotted line, Fig. III. This is given by the full lines in the present figure and is deduced, in the manner already explained, from the turning moments, Fig. III, and the reaction at bearing A, Fig. VII. The dotted lines give the steam turning moments when bearing B is removed, deduced similarly from the reactions for this case, Fig. XXIV.. The figure is constructed in the same way as Fig. III; positive turning moments for the 1st order of cranks being laid off below AB, and for the 2d order above AB.

Fig. XXIV gives the reactions (induced by steam forces F_1, F_2 and F_3), when bearing B is removed, deduced by means of equation (16). These curves were shown partially in Fig. VII, but are here given complete; the present figure being similar in construction to the earlier one. For convenience the full line X—X has been added giving the reaction on the crank pin—from Fig. II—laid off on the crank pin path expanded, as base.

In Fig. XXV is shown the reactions at bearing A due to inertia when bearing B is removed, deduced by means of equation (16). The figure is similar to Fig. XII.

An examination of these figures shows that, as before, the parts of the revolution at which maximum stresses may occur are when the L.P. crank is approaching top center and when it is just past bottom center, for either order of cranks. Calculation of these four cases shows that the only one for which the stresses need be tabulated here is when the L.P. crank is approaching top center and the cranks are in the 2d order. The results in the other cases will merely be referred to where necessary.

Tables XXI and XXII give the estimation of the moments due to steam and inertia, respectively, required for this case. Only the following need be noted:—

Lines (d). The turning moment for the forward web, column VII, is measured near the crank pin end, as there the inertia turning moment is greater than near the shaft center.

Line (f), Table XXI. The multiplier for the increase due to unequal steam stresses is applied to the turning moments in columns V, VII and X. This is usually the worst case as the steam bending moments are now comparatively small. It is not perfectly consistent with the corresponding change in the moments stressing the crank pin, Table XIII; but, except for the small values of b, the difference is exceedingly slight.

Lines (g) of table XIV and XV. These are omitted here for the same reason as was noted in connection with Table IX.

Lines (k). There is no allowance for change of section in columns III, IV, VII, VIII and IX.

In Table XXIII are given the values of the forces F_1, F_2 and F_3. From these the deflection, Y_B, at B is calculated by the equation

$$Y_B = \frac{1.768 \times 10^{-9}\, b^3}{a^4} (-6F_1 + 23F_2 + 29F_3); \quad . \quad (24)$$

which, remembering the convention with regard to positive forces, gives the deflection positive when the shaft moves down.

Before being entered in Table XXV each of the values of Y_B was multiplied by its factor for the influence of the webs. (It should have been noted earlier that these factors will be found in one of the appendices.) As we will require the deflection for both orders of cranks, both sets of forces are given in Table XXIII. The forces due to steam can at once be read from Fig. XXIV; those due to inertia can readily be estimated from Fig. X, using the division of moving masses given in Table XII. In line (g), the L.P. steam forces are supposed to be ten per cent. higher than would be due to equal distribution. This is the most likely supposition and will increase the deflection. The

changes supposed for the H.P. and I.P. are also such as to increase the deflection.

Table XXIV gives the reaction at bearing A when bearing B is removed.

In Table XXV are given the stresses, so far as seemed necessary, calculated by means of Tables XXI, XXII and XXIII. No stresses less than 15,000 pounds per square inch have been tabulated. The stresses in this table being for the 2d order of cranks, while Tables XIII and XIV were for the 1st order, the rise of stress is not given by the excess of these figures over 15,000; and the omission of lines (g) from Tables XXI and XXII, which seems entirely justified, makes the present stresses smaller than they would otherwise be. Indeed, the stresses calculated for the 1st order of cranks with the L.P. crank near top center, in some cases fall below 15,000 on account of this omission. These facts partly account for an apparent discrepancy between the stresses in Tables XIX and XXV, those in the latter table being smaller than might be anticipated. Another cause of apparent discrepancy will be noted further on.

Lines (a). Only when the stress in the aft L.P. journal is greater than in the forward has it been noted. For $F_m = 10$ and 20 the stresses are higher for the 1st order of cranks and the same position of the L.P.; but, as they are then just about 15,000 pounds per square inch, they have not been tabulated. When the L.P. crank is 27° past bottom center the stress is always smaller than for the other cases.

Lines (b). In all cases where the moments of Tables XXI and XXII give the stress in the after L.P. web greater than 15,000 pounds per square inch, it is smaller than the corresponding stress in the forward L.P. web; thus all values are excluded from these lines.

Lines (c). For $\mathfrak{M} = 10$ and the L.P. crank near top center the stresses for the 2d order of cranks are rather larger than for the 1st order from $F_m = 100$ down to $F_m =$ a little larger than 40, below which value the 1st order gives the larger stress. Thus the stress 15,663 in line (c_4) is not quite the maximum, though

within one per cent. of it. For $\mathfrak{M}$ less than 10 the ratio of the stresses given to those for the 1st order of cranks is greater for the higher values of F_m than when $\mathfrak{M} = 10$, and the value of F_m when these stresses are equal is lower the lower the value of $\mathfrak{M}$. The stress in the crank pin for the L.P. crank 27° past bottom center is never important.

Lines (d). The results are similar to those in lines (c); only, the ratios of the stresses given to those for the 1st order of cranks when the L.P. crank is approaching top center, is greater in the present case. For the higher values of F_m the ratio is still very near unity.

Lines (e). Comparing again the 1st and 2d order of cranks when the L.P. crank is approaching top center, the stress is always greater for the case given in Table XXV. For the values above 15,000 and $\mathfrak{M} = 10$, the stress for the 1st order is from 7 to 10 per cent. lower than that for the 2d—a larger percentage than for lines (c) and (d). For lower values of $\mathfrak{M}$ this percentage is greater.

When the cranks are in the 2d order and the L.P. 27° past bottom center, the stresses for the higher values of F_m are lower than those given in the table; but for $\mathfrak{M} = 10$ equality is reached when F_m is a little over 40. For $F_m = 40$ the value now is 15,463. The smaller $\mathfrak{M}$ is the lower is the value of F_m at which this equality occurs. The case of the 1st order of cranks and the L.P. 27° past bottom center is never important.

Lines (f). These deflections are in thousandths of an inch per one inch stroke of engine. Being for the 2d order of cranks they correspond to the values in the preceding lines of the table. All these deflections are negative, though written without a sign; that is, the shaft is deflected upward, and thus the previous lines give the stresses actually occurring on removal of the main bearing cap B of an inverted vertical engine.

Lines (g). These are the deflections for the 1st order of cranks and always correspond to a somewhat lower stress than is given in the preceding lines. They are also negative.

It may not be quite evident why in Table XIX, lines (f),

the case tabulated as the worst for the forward L.P. main journal is that of the 1st order of cranks and the L. P. crank 27° past bottom center, while here we find the highest stress is for the 2d order of cranks, and the L.P. about 27° before top center; but, considering both cases with the bearings in line, the ratio of the turning to the bending moment is greater for the former case. Consequently, while the stresses for the shaft in line are then greater, the rate of rise of stress per .001 inch error of bearing B is greater for the latter case; and, when bearing B is removed, the deflections at B are considerably the greater in the latter case, for the higher values of F_m. Thus, this apparent discrepancy is completely explained.

Table XXVI gives the nominal pressure in pounds per square inch on bearing A when bearing B is removed, calculated from Table XXIV. This is most readily done by increasing the pressure for the shaft in line—400 pounds per square inch—in the ratio of $.295F_m + .617G$ (Table XV), to $.479F_m + .751G$ (Table XXIV). Line (a), Table XXVI, gives the result while line (b) gives the value of the above ratio. The increase is seen to depend only on the value of G / F_m or $\mathfrak{M}$. Though the nominal pressures on the crank pins are not affected by removing bearing B, the distribution of pressure will not be so good as with the bearings in line on either these or the main bearings, on account of the increased flexure of the shaft.

The principal conclusions which flow from what immediately precedes are:—

1st. Rise of stress when bearing B is removed is considerable only for the higher values of F_m. Then the stress is always greatest when the L.P. crank is about 27° before top center.

2d. The stress is greater the greater the values of F_m and $\mathfrak{M}$.

3d. While the greatest stress occurs for the 2d order of cranks, nearly as high stresses are produced when the cranks are in the 1st order, and that for smaller deflections at B. Consequently, we cover all cases with sufficient accuracy by taking the stresses in lines (c), (d) and (e) along with the deflections in lines (g).

4th. The deflections—the actual values of which are propor-

tional to the stroke of the engine—rise with increase of F_m and $\mathfrak{M}$. They are such that, in an engine of even moderate stroke and a fairly high value of F_m and $\mathfrak{M}$, the shaft would not run quietly if the cap of bearing B was removed.

Consequently, for such engines, the elasticity of the shaft forms a safeguard against rise of stress from excessive easing of the main bearing caps; and, if the bearings are nearly in line, the stresses will rarely or never rise to the values given in Table XXV. The easing of a bearing cap by a mere fraction of a hundredth of an inch (actual rise of cap, not always to be measured by the unscrewing of the nuts, as the main bearing bolts &c., may be under considerable stress which has to be relieved before the cap begins to rise) will often produce a sufficient improvement in the lubrication to stop heating. But in all cases Table XXV gives the limiting rise of stress from slacking the main bearing bolts. Removing two adjoining bearings at one time would give such excessive deflections of the shaft that this condition is at once seen to be beyond the limits which call for practical consideration.

But if the bearings of the engines referred to in the last paragraph have from any cause got badly out of line, this elasticity of their shafts still limits the amount of slacking back of the main bearing bolts, on account of the excessive knocking which is at once produced. Thus, there is no reason for diminished vigilance in watching the alignment of the bearings. In this case the stresses which may be induced can most readily be calculated from Table XIX. However, it should be carefully remembered that that table gives the worst set of conditions and the very greatest rate of rise of stress which need be considered.

The figures in Table XXV and many preceding tables must not be taken too absolutely. They have been got under restrictions which I have endeavored in all cases to explain clearly, and will be somewhat modified by the variations of these restrictions which occur in practice. I think, however, that a careful examination will lead to the conclusion that the figures given are always amply approximate for our purpose. In this connection it may be explained that all figures given have been care-

fully checked by a second computation and, where they formed a series, curves representing the figures and their first differences have in almost all cases been plotted. Where any error was shown by these curves the calculations were corrected, except when the error corresponded to a small fraction of one per cent. The calculations were made principally by means of Fuller's Spiral Slide Rule, which greatly shortens the time and lightens the labor required for such an investigation. Indeed without some such instrument the work could hardly have been undertaken by a single individual.

From Table XXV we can readily ascertain approximately the values of the stress in the crank pin, forward web, and main journal of the L.P. shaft, corresponding to the values of F_m and $\mathfrak{M}$ for the ten reference engines, Tables IV and XI. Judging the allowable stress from the mean and maximum values of such a table, in the same way as was done for the shaft in line, justifies only a slight increase over 15,000 pounds per square inch. (This procedure is not quite similar to that adopted when steam stresses alone were being dealt with, where all the figures in lines (c), (d) and (e), Table X, were used. But the range of values of $\mathfrak{M}$ in the ten reference engines is so restricted that the procedure now adopted seems the only fair one.) This increased stress has been taken at 15,500 pounds per square inch.

To find the dimensions of shafts which, for bearing B removed, will have this stress as a maximum I have followed the earlier procedure. The stresses in the crank pin, forward L.P. web, and main journal (partly given in lines (c), (d) and (e), Table XXV), were divided by 15,500 and the cube root of the ratio taken. This gives multipliers which, applied to the crank pin and main journal diameters, and web breadth and thickness, Figs. XIV to XXI, give the required dimensions—neglecting, as before, the slight change in b. The multipliers so obtained were for $\mathfrak{M} =$ 2, 5, 8 and 10; and $F_m = 10$, 20, 40, 60, 80 and 100. The completion of the series was got by interpolation and hence the tables subjoined to Figs. XIV, XVI, XVIII and XIX. Multipliers less than unity have not been retained; and it was not considered necessary to give tables for the consequent changes

in Figs. XX and XXI, the ordinates of these curves being changed by twice the addition to the thickness of web.

The following suggestions are made regarding the use of what precedes:—

For 60,000 pound steel no reduction should be made in any part of the shaft from the ordinates of the curves. If the deflection at B with that bearing removed is about 1/40 inch, the ordinates of the curves, increased by the corresponding multipliers in the case of Figs. XIV, XVI, XVIII and XIX, should be used. If the deflection is greater, somewhat smaller multipliers may be adopted; till, at a deflection of 1/10 inch, they might very safely be omitted.

If the deflection at B is much less than 1/40 inch, it would be well to make a greater allowance than the multipliers give; as, with all the bearing caps screwed down, the amount by which the bearings have to be out of line to raise the stresses above those given in Table XXV is so minute as to make the occurrence of these higher stresses at some time probable. The dimensions which will give any chosen stress under definite conditions of error of bearing can readily be calculated with sufficient exactitude from Table XIX. To make the matter clear I will give an example.

Example IV.—Suppose an engine with $F_m = 60$, $\mathfrak{M} = 5$ and the stroke $= 15$ inches.

The deflection at B from Table XXV, with that bearing removed, $= .852 \times 15 \times 10^{-3} = .0128$, or about 1/80 inch. As this is under the above limit at which it is proposed to use the multipliers, let us determine what shaft dimensions will be required to give, say, 16,000 pounds per square inch—that is, to leave a factor of safety of about two from the elastic limit—when the shaft at B is held out of line by 1/50 inch. This is $\frac{1}{50 \times 15} = \frac{1}{750}$ inch per one inch stroke. The deflection is not so great as to realize the extreme conditions of Table XX.

We must now recollect that in changing the shaft diameter little change will be made in the rise of stress due to want of alignment. We then have:—

TABLE XXVII.—ESTIMATION OF MULTIPLIERS—EXAMPLE IV.

Line.	Part of shaft.	L.P. crank pin.	L.P. forward web.	L.P. main journal.
(a)	Stress for shaft in line. Table XIX, lines (d'_1), (e'_1) and (f'_1).....	15,000	12,444	12,543
(b)	Rise of stress per .001 inch error of bearing *B*. Table XIX, lines (d_1), (e_1) and (f_1).....	2,576	4,913	4,452
(c)	Rise of stress per $\frac{1}{750}$ inch error of bearing *B*	3.435	6,551	5,936
(d)	Total stress in altered shaft	16,000	16,000	16,000
(e)	Stress in altered shaft for bearings in line. Line (d) — line (c).....	12.565	9,449	10,064
(f)	Stress in line (a) ÷ stress in line (e).....	1.194	1.317	1.246
(g)	Required multipliers $= \sqrt[3]{\text{line (f)}}$.....	1.061	1.096	1.076

If the length of the bearings are left the same, we have :—

Thickness of crank web, Fig. XIX, . = .248 per in. stroke.
Increase for two webs, line (g), Table XXVII = .248 × 2 × .096 = .048 " " "
Span of main bearings, Fig. XX, = 1.425 " " "
Increased span of main bearings, = 1.473 " " "
Proportionate increase of span, = 1.473 / 1.425 = 1.034

This very appropriately measures the increase of bending moments from steam and inertia, and the stress for the shaft in line is less than 3 per cent. over that estimated. The change in rise of stress from want of alignment, measured from the change in the ratio a/b^2, $1.076/1.034^2 = 1.006$. Thus the method used in Table XXVII is almost exact in this case, and will always be so unless b is much changed. In such cases the degree of accuracy can be at once tested, as just shown.

Let us now examine the corrections which are required in the dimensions of the shaft for a change in the span of the main bearings, &c. As before, I will only take the case of the bearing *B* removed, and neglect the slight change of pressure on the main bearings.

The following values will occur :—

TABLE XXVIII.

TABLE OF CONSTANTS FOR CORRECTING SHAFT DIMENSIONS.

$$C = .275\ (1 + .206\ \mathfrak{M}).$$
$$J = .504\ (1 + .414\ \mathfrak{M}).$$
$$J_1 = .462\ (1 + .767\ \mathfrak{M}).$$
$$W = .392\ (1 + .333\ \mathfrak{M}).$$
$$W_1 = .387\ (1 + .578\ \mathfrak{M}).$$
$$W_T = .578\ (1 + .0207\ \mathfrak{M}).$$

These are constant for similar engines with the same steam pressure and piston speed.

Crank Pin.—

Turning moment $= .826\ F_m + .0955\ G$ (Tables. XXI and XXII.)
$= .826\ F_m + .0955 \times .2044\ \mathfrak{M}\ F_m$
$= .826\ (1 + .0236\ \mathfrak{M})\ F_m = \mathfrak{T}\ F_m$

Bending moment $= .196\ F_m b + .320\ Gb$ (Tables XXI and XXII.)
$= .196\ (1 + .334\ \mathfrak{M})\ F_m b = \mathfrak{B}\ F_m b.$

Equivalent augmented turning moment

$$= [\mathfrak{B}b + \sqrt{\{\mathfrak{B}^2 b^2 + \mathfrak{T}^2\}}]\ F_m.$$

If b is increased to $b + \delta b$, we have for the corresponding change of crank pin diameter from γ to $\gamma + \delta\gamma$,

$$\gamma^3 : (\gamma + \delta\gamma)^3 ::$$

$$\mathfrak{B}b + \sqrt{\{\mathfrak{B}^2 b^2 + \mathfrak{T}^2\}} : \mathfrak{B}\ (b + \delta b) + \sqrt{\{\mathfrak{B}^2\ (b + \delta b)^2 + \mathfrak{T}^2\}}.$$

If we put $\mathfrak{R}^2 = \mathfrak{B}^2 b^2 + \mathfrak{T}^2$, we get

$$1 : \left(1 + \frac{\delta\gamma}{\gamma}\right)^3 :: \frac{\mathfrak{B}b}{\mathfrak{R}} + 1 : \frac{\mathfrak{B}b}{\mathfrak{R}} + \frac{\mathfrak{B}\delta b}{\mathfrak{R}} + \sqrt{\left\{1 + \frac{2\mathfrak{B}^2 b\delta b}{\mathfrak{R}^2}\right\}},$$

neglecting the square of δb, as it will never be more than a small fraction of b. Expanding, retaining only two terms of the expansions, and transforming, we get,

$$1 : \frac{3\delta\gamma}{\gamma} :: \frac{\mathfrak{B}b}{\mathfrak{R}} + 1 : \frac{\mathfrak{B}\delta b}{\mathfrak{R}} + \frac{\mathfrak{B}^2 b\delta b}{\mathfrak{R}^2} :: \frac{\mathfrak{B}b}{\mathfrak{R}} + 1 : \frac{\mathfrak{B}\delta b}{\mathfrak{R}}\left(\frac{\mathfrak{B}b}{\mathfrak{R}} + 1\right) :: 1 : \frac{\mathfrak{B}\delta b}{\mathfrak{R}};$$

and, $$\delta\gamma = \frac{\mathfrak{B}_\gamma \delta b}{3\mathfrak{R}} = \frac{\mathfrak{B}_\gamma \delta b}{3\sqrt{\{\mathfrak{B}^2 b^2 + \mathfrak{T}^2\}}} = \frac{\mathfrak{B}_\gamma \delta b}{3\mathfrak{T}\sqrt{\left\{1 + \frac{\mathfrak{B}^2 b^2}{\mathfrak{T}^2}\right\}}}$$

$$= \frac{.196\,(1 + .334\ \mathfrak{M})\,\gamma\delta b}{3 \times .826\,(1 + .0236\ \mathfrak{M})\sqrt{\left[1 + \left\{\frac{.196\,(1 + .334\ \mathfrak{M})}{.826\,(1 + .0236\ \mathfrak{M})}\right\}^2 b^2\right]}}$$

It will be found by trial that, between the limits $\mathfrak{M} = 2$ and $\mathfrak{M} = 10$, we have very approximately

$$\frac{.196\,(1 + .334\ \mathfrak{M})}{.826\,(1 + .0236\ \mathfrak{M})} = .275\ (1 + .206\ \mathfrak{M}) = C \text{ of Table XXVIII.}$$

Thus we have finally :—

For one inch stroke, $$\delta\gamma = \frac{C_\gamma \delta b}{3\sqrt{\{1 + C^2 b^2\}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

For stroke s, $$\delta\gamma' = \frac{C_\gamma' \delta b'}{3\sqrt{\{s^2 + C^2 b'^2\}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

Main Journal.—The corresponding expressions for the addition $\delta\alpha$ to the main journal, are :—

For one inch stroke, $$\delta\alpha = \frac{(J\,\delta H - J_1\,\delta H_1)\,\alpha}{3\sqrt{\{1 + (JH - J_1 H_1)^2\}}} \quad . \quad . \quad . \quad (27)$$

For stroke s, $$\delta\alpha' = \frac{(J\,\delta H' - J_1\,\delta H'_1)\,\alpha'}{3\sqrt{\{s^2 + (JH' - J_1 H'_1)^2\}}} \quad . \quad . \quad (28)$$

J and J_1 are as given in Table XXVIII.

Crank Web.—For a stress of 15,500 pounds per square inch, the breadth ε and thickness e must satisfy the following equations :—

For one inch stroke,

$$\frac{F_m}{\varepsilon e}\left\{\frac{W_T}{\varepsilon} + \frac{WK - W_1 K_1}{e}\right\} = 2{,}583. \quad . \quad . \quad (29)$$

For stroke s,

$$\frac{F_m s^2}{\varepsilon' e'}\left\{\frac{W_T s}{\varepsilon'} + \frac{WK' - W_1 K'_1}{e'}\right\} = 2{,}583. \quad . \quad . \quad (30)$$

W_T, W and W_1 are as given in Table XXVIII.

As the areas of the main bearing and crank pin, given by Figs. XIV to XXI, correspond to pressures of 400 and 600 pounds per square inch, respectively, with all bearings in line (the pressures for the main bearing A rising to the values given in Table XXVI, if bearing B is removed) the area required for any desired pressure can at once be calculated.

Example V.—Let us find by means of equations (25) to (30) what changes would be required in the crank shaft of *Example III* if the main bearing A was shifted 5 inches aft and the surface pressure of the main bearings was reduced to 330 pounds per square inch. For comparison, the change of the main bearing is the same as in *Example II* and the reduction of bearing pressure is nearly proportionate.

It is found from Table XXV that the deflection of this shaft at B, when that bearing is removed, is about $1.7 \times 10^{-3} \times 30 = .051$. As this deflection is comparatively large, following the suggestions made for the use of the multipliers of Figs. XIV, XVI, XVIII and XIX, would change the ordinates of the curves in all cases by less than one per cent. This change is neglected here.

The following constants from Table XXVIII will be required:—

$$\begin{aligned} C &= .275\,(1 + .206 \times 7.6) = .706. \\ J &= .504\,(1 + .414 \times 7.6) = 2.090. \\ J_1 &= .462\,(1 + .767 \times 7.6) = 3.155. \\ W &= .392\,(1 + .333 \times 7.6) = 1.384. \\ W_1 &= .387\,(1 + .578 \times 7.6) = 2.087. \\ W_T &= .578\,(1 + .0207 \times 7.6) = .669. \end{aligned}$$

Neglecting the change of main journal diameter, the main bearing will have to be lengthened to

$$\frac{16.53 \times 400}{330} = 20.04 \text{ inches.}$$

Crank pin.—b' will be increased from 52.23 to

$$52.23 + 5 + 20.04 - 16.53 = 60.74.$$

Thus $\delta b' = 60.74 - 52.23 = 8.51.$

From equation (26)

$$\delta\gamma' = \frac{.706 \times 11.76 \times 8.51}{3\sqrt{\{30^2 + .706^2 \times 52.23^2\}}} = \frac{.706 \times 11.76 \times 8.51}{3 \times 47.54} = .50,$$

and the new diameter of crank pin is

$$11.76 + .50 = 12.26 \text{ inches.}$$

Main Journal.—We have,

$$H' = \tfrac{1}{2}(52.23 + 18.39) + 8.10 = 43.41.$$

$$H'_1 = \frac{18.39}{2} + 8.10 = 17.30.$$

$$\delta H' = 5 + \tfrac{1}{2}(20.04 - 16.53) = 6.75.*$$

$$\delta H'_1 = 0.$$

From equation (28)

$$\delta\alpha' = \frac{2.09 \times 6.75 \times 10.92}{3\sqrt{\{30^2 + (2.09 \times 43.41 - 3.155 \times 17.30)^2\}}}$$

$$= \frac{2.09 \times 6.75 \times 10.92}{3 \times 46.97} = 1.09,$$

and the new diameter of shaft is

$$10.92 + 1.09 = 12.01 \text{ inches.}$$

Crank Webs.—As before, ε', the fore and aft thickness of the webs, is supposed to remain the same. The breadth, ε', must then be determined to keep up the strength.

We have now,

$$K' = \frac{20.04}{2} + .55 + 5 + 8.10 \times 1.5 + 18.39 = 46.11,$$

$$\text{and } K'_1 = \tfrac{1}{2}(18.39 + 8.10) = 13.25.$$

From equation (30) we get,

$$\frac{64.11 \times 30^2}{8.10 \times \varepsilon'}\left\{\frac{.669 \times 30}{\varepsilon'} + \frac{1.384 \times 46.11 - 2.087 \times 13.25}{8.10}\right\} = 2,583.$$

* In Vol. VIII, page 723, we should have had

$$\delta H' = 6.48,$$

$$\delta\alpha' = .62, \text{ and}$$

new diameter of shaft = 10.97 inches.

The positive root of this equation gives

$$\varepsilon' = 15.81 \text{ inches.}$$

The same lesson may be drawn from these results as flowed from the comparison of *Examples I and II.*

We may also use the curves of Figs. XIV to XXI to determine the size of shaft which will have any desired stress. Suppose we wish to raise the stress for a particular shaft when in line from 15,000 to 18,000 pounds per square inch. This will occur simultaneously in the aft main journal, aft web, and crank pin, only if F_m and G are both raised in this proportion—that is, if F_m is raised in this proportion while G / F_m, or $\mathfrak{M}$, is constant. Thus if the engine has $F_m = 60$ and $\mathfrak{M} = 7$ we must determine the shaft from Figs. XIV to XXI for $F_m = \frac{60 \times 15}{18} = 50$ and $\mathfrak{M} = 7$. The pressure on the aft main journal and L.P. crank pin will now be $\frac{400 \times 18}{15} = 480$ and $\frac{600 \times 18}{15} = 720$ pounds per square inch respectively.

Changes in shaft dimensions for a change of b, &c., would still be determined from the formulas given above; only, in equations (29) and (30), the value of F_m to be used is 50—not 60. If it is desired to run the main bearing at 420, and the crank pin at 660 pounds per square inch, respectively, it will obviously be necessary to determine the alteration of bearing area to give pressures of

$$\frac{420 \times 50}{60} = 350 \text{ and } \frac{660 \times 50}{60} = 550$$

pounds per square inch, when $F_m = 50$.

This concludes the investigation for the shaft with three cranks at 120°, excepting a considerable amount of matter not necessary to the argument but which, still, is of value. This will be given in the appendices.

The series of crank shafts chosen is one of the three most commonly occurring in practice. In treating it, only important elements and those minor ones which could be included readily have been used. The stresses omitted—such as the shearing

stress on the main journal—would have but slightly affected the final result while greatly complicating the problem; but their values have been estimated to determine if in any case they rose to importance. The treatment of the effect of change of section, by factors in the tables of moments, can only be looked on as superficial; but the elastical problem involved must be one of great difficulty, while the small factors introduced probably cover this effect where a reasonably large fillet is made.

The result of the whole investigation brings out most forcibly the enormous importance of inertia as a factor. This is clearly shown by Figs. XIV and XVI, especially if we remember that the strength of the sections varies as the cubes of the ordinates of these curves. The importance of properly taking account of the bending moment is also emphasized; and the use of formulas of the ordinary types is only partially justified for engines of very moderate piston speed. During the last fifteen or twenty years piston speeds have risen very considerably, and in many cases now occurring any formula which fails to take account of inertia cannot be considered even an approximate solution of the crank shaft problem. No doubt the constants in the present formulas have been placed high enough to cover with reasonable safety the large proportion of the engines being built; but if so, there must be a want of economy in the slower speed engines, and these rules tend to the adoption of too light shafts when the usual limits of speed are much exceeded. On the other hand the adoption of correct rules tends both to uniformity of the factor of safety and, in all cases, to the greatest economy which can be properly attained. Very possibly, by a more comprehensive discussion of present practice it might be found safe to raise considerably the maximum stresses which have been adopted throughout this investigation.

Part of the object in presenting the investigation so fully was to exhibit the effect of all the important factors which entered into it. But for practical use further simplification, if desired, might be attained by once for all making some of the allowances which are here given separately. For instance, the curves might be changed sufficiently in the way indicated by the multipliers at-

tached to Figs. XIV, XVI, XVIII and XIX, and a partial allowance made for such changes as are contemplated in equations (25) to (30); so that these multipliers and equations might be omitted, at least within large limits, without introducing too great inaccuracy. Such changes should, however, only be made after very careful discussion.

That such investigations should be extended to cover the several series of crank shafts in ordinary use does not, I think, admit of doubt, notwithstanding the very large amount of labor which would be involved. This might proceed on a similar plan to that followed here; or, where the proportionate changes in the moments and reactions are not great, one series might more easily be derived by approximations from a preceding one. The artifices to be employed in each case would, no doubt, readily suggest themselves. Much work given above and to be given in the appendices, is immediately applicable; and it is hoped that the very full discussion in the present case will, by the light thrown on the whole question, make it possible to greatly shorten succeeding investigations.

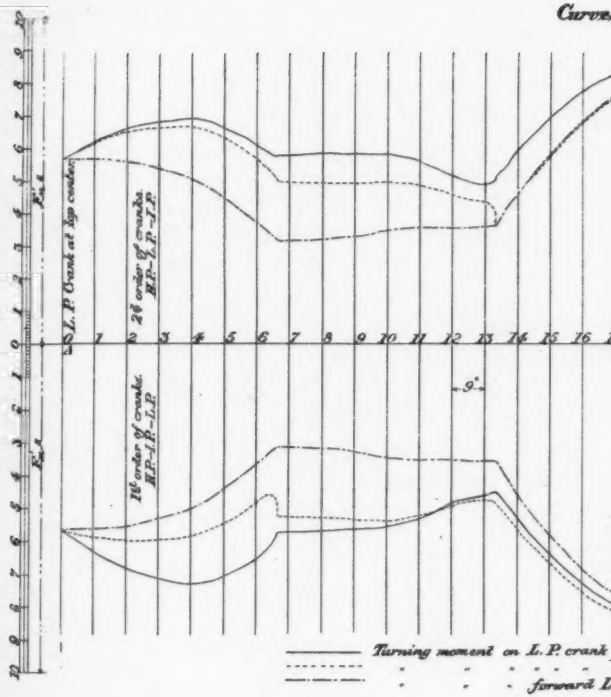
It might be possible, after such full investigation, to combine all solutions in one series of curves or in one set of tables. If, for instance, the shaft dimensions were given by a series of curves the abscissæ of which were values of a simple function depending on F_m, and also partly on the particular type of crank shaft, instead of being equal to F_m—as in the preceding figures; and each individual curve, instead of being for a constant value of $\mathfrak{M}$, was or a constant value of a simple function depending on $\mathfrak{M}$ and the particular type of crank shaft; the several series of curves or tables might be forced to agree with sufficient accuracy, though simplicity does not depend to any important extent on such a reduction. Where an engine has to conform to the rules of a Supervising Board, formulas or tables only can be used, as curves always involve more or less inaccuracy in reading, on which errors important questions might depend.

Careful thought will, I think, make it appear at least possible to evolve a system of rules, covering the whole range of practice, quite as simple and convenient as those at present in use;

excepting, of course, the additional but indispensable trouble of estimating the moving masses. How far it would be wise to advocate the adoption of such a set of rules, however simple, as compulsory throughout the merchant marine, is open to grave consideration, and could not be discussed profitably at this stage. All that is connected with the speed of engine and mass of the moving parts would then have to come, to some extent, under the control of the Supervising Boards, and this would, unless great care was exercised, lead to the framing of too rigid and minute rules. These might not only hamper the development of good design, but would tend to relieve the designer of responsibility which is properly his—a responsibility which is, and happily must be, left to him in many cases of quite as vital importance as that dealt with here.

Finally, whatever system is adopted it must fulfill the three conditions—simplicity, fair accuracy, and sufficient elasticity to give free scope to the development of good design.

Curve


L.P. Crank at top centre
2nd order of cranks
H.P.–L.P.–I.P.
1st order of cranks
H.P.–I.P.–L.P.
9°
Turning moment on L.P. crank
forward

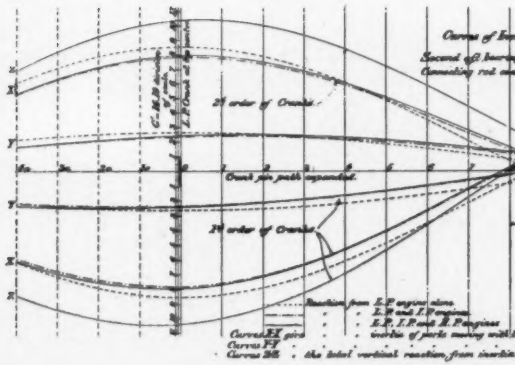
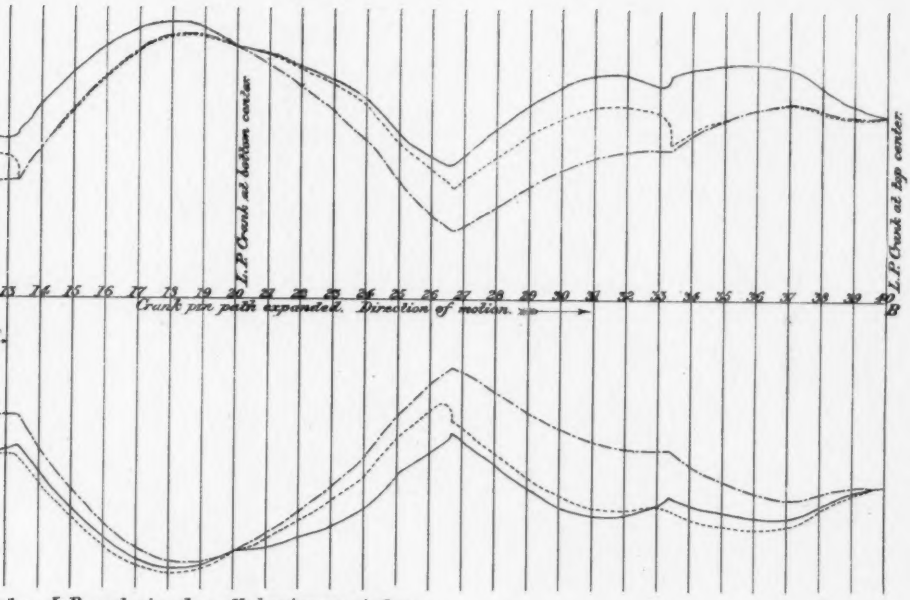

2nd order of Cranks
1st order of Cranks
Crank pin path expanded.
Reaction from L.P engine alone
Curves EF give
Curves FF
Curves BE

FIG. XXIII.

Curves of Steam Turning Moments.



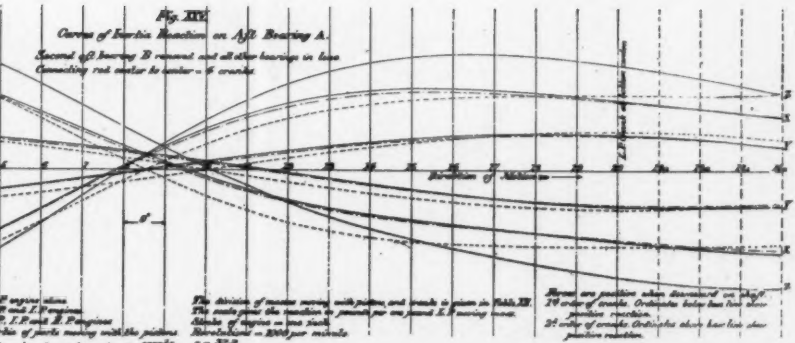
...at on L.P. crank pin when all bearings are in line.
... " " " " " The second aft bearing, B, is removed.
... forward L.P. main journal, from Fig. III.

Fig. XIV.

Curves of Inertia Reaction on Aft Bearing A.

Second aft bearing B removed and all other bearings in line.

Connecting rod under to under = 6 cranks.



...engines alone.
...and L.P. engines.
...H.P. and L.P. Engines
...ion of parts moving with the pistons
...crank.
...al reaction from inertia.

The division of reaction moving with pistons and cranks is given in Table XII.
The curves show the reaction in pounds per sq. inch of L.P. moving piston.
Revolutions = 100 per minute.
$g = 32.2$.

Thrust are positive when downward on shaft.
1st order of cranks. Ordinates below base line show positive reaction.
2d order of cranks. Ordinates above base line show positive reaction.

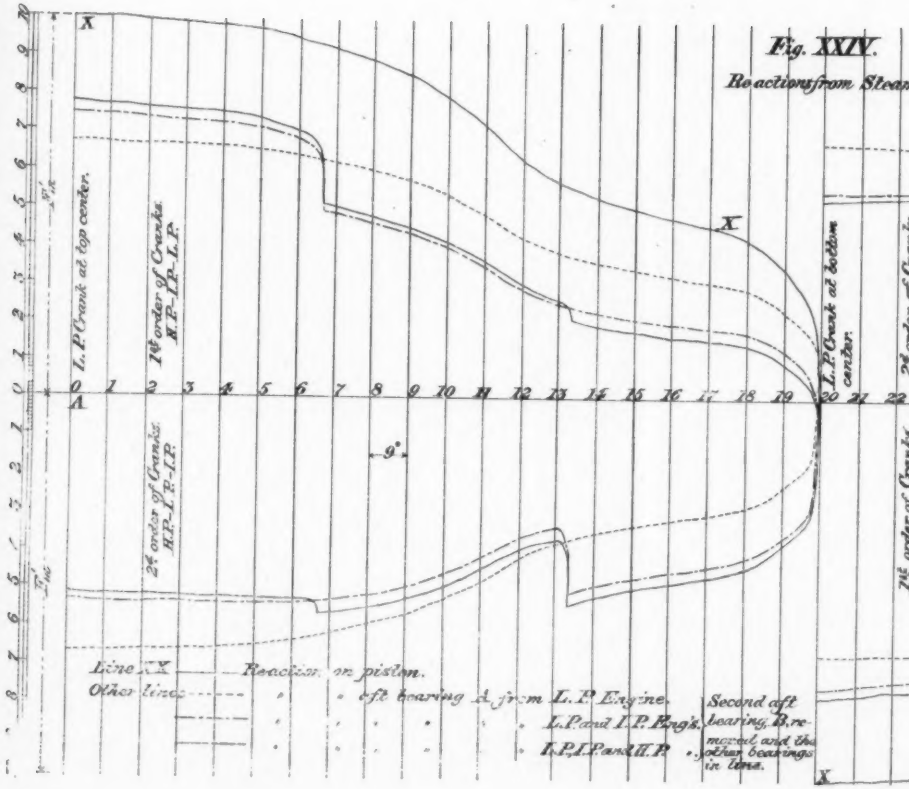


Fig. XXIV.
Reactions from Steam
X
X
L. P. Crank at top center.
1st order of Cranks H.P.-L.P.-L.P.
2d order of Cranks H.P.-L.P.-I.P.
L. P. Crank at bottom center.
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22
A
9°
Line X X —— Reaction on piston.
Other lines ---- " " " aft bearing from L. P. Engine.
" " " " " L. P. and I. P. Eng's.
" " " " " I. P. I. P. and H. P.
Second aft bearing, B, removed, and the other bearings in line.
X

Fig. XXIV.

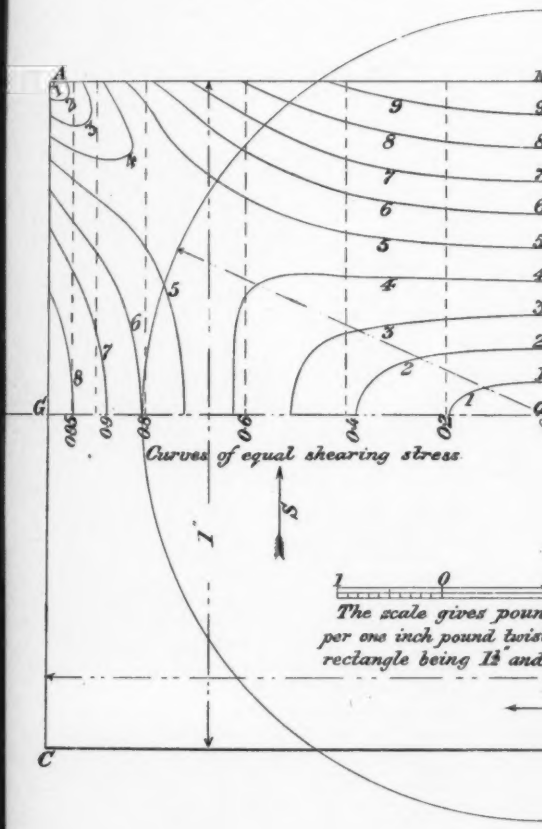
...ions from Steam.

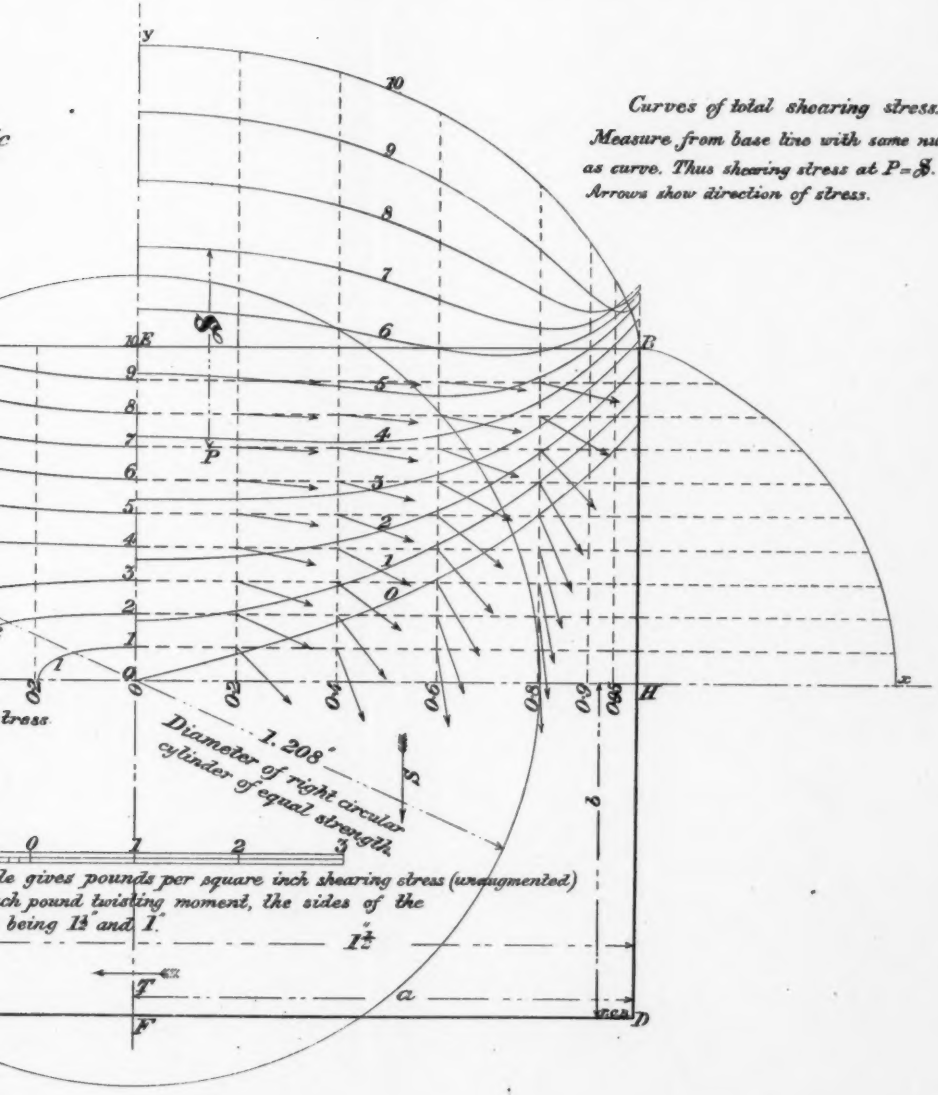
L.P. Crank at bottom center.
2d order of Cranks. H.P.-L.P.-L.P.
L.P. Crank at top center.
20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40
Crank pin path expanded. Direction of motion.
B
2d order of Cranks. H.P.-L.P.-L.P.
X
X

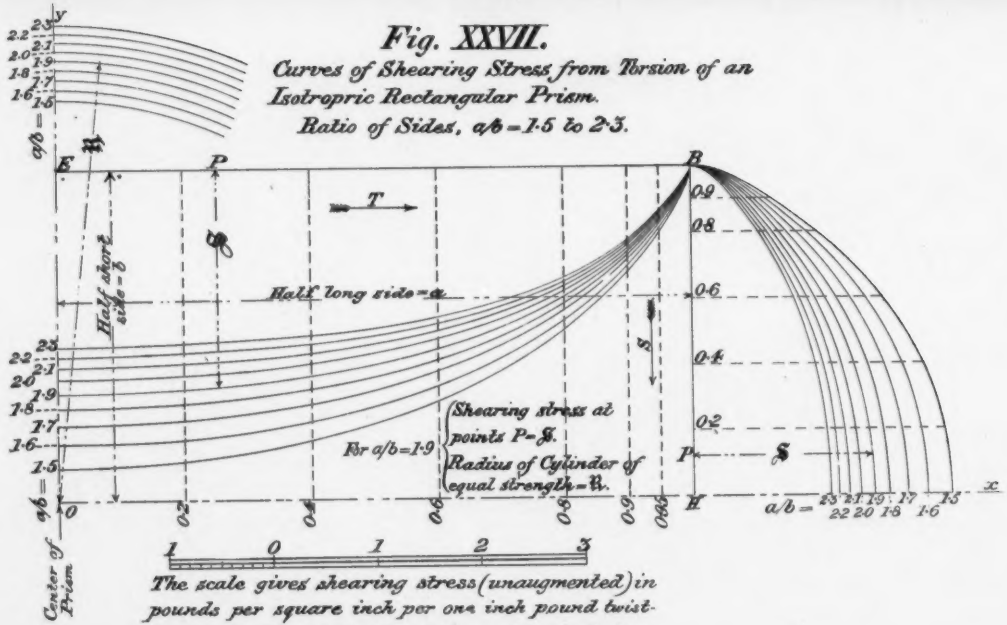
Fig. XXVI.

Curves of Shearing Stress from Torsion of an Isotropic Rectangular Prism.

Ratio of Sides, a/b = 1·5.




Curves of total shearing stress.
Measure from base line with same number as curve. Thus shearing stress at P=8.
Arrows show direction of stress.
y
x
E
B
H
D
F
P
T
a
b
S
Diameter of right circular cylinder of equal strength
1.208"
1½"
gives pounds per square inch shearing stress (unaugmented)
pound twisting moment, the sides of the
being 1½" and 1"



The scale gives shearing stress (unaugmented) in pounds per square inch per one inch pound twisting moment, the short side of the rectangle being one inch and the ratio of sides, a/b, as marked.

TABLE XIX.—EFFECT OF BEARING *B* BEING .001 INCH OUT OF LINE.

The other bearings are supposed to be in line. Stroke of engine = 1 inch.

Line.			*Fm.*	10	20	40	60	80	100
(a_1)	$\frac{B}{m}=5$,		Increased reaction at bearing *A*	509	305	177	129	103	86
(b_1)		Increase of maximum stress.	Aft L. P. main journal	1,033	949	842	783	741	709
(c_1)			Aft L. P. web	2,964*	2,402	1,873	1,607	1,439	1,311
(d_1)			L. P. crank pin	1,515† / 4,677‡	1,979 / 3,864	2,064 / 3,003	1,979 / 2,576	1,872 / 2,298	1,758 / 2,081
(e_1)			Forward L. P. web	9,147*	7,445	5,794	4,913	4,335	4,057
(f_1)			Forward L. P. main journal	7,600	6,287	5,061	4,452	4,051	3,765
(b'_1)		Maximum stress when all bearings are in line	Aft L. P. main journal	15,000	15,000	15,000	15,000	15,000	15,000
(c'_1)			Aft L. P. web	13,636*	13,636	13,636	13,636	13,636	13,636
(d'_1)			L. P. crank pin	15,000	15,000	15,000	15,000	15,000	15,000
(e'_1)			Forward L. P. web	12,035*	12,155	12,350	12,444	12,560	12,920
(f'_1)			Forward L. P. main journal	12,736	12,708	12,632	12,543	12,458	12,400
(a_2)	$\frac{B}{m}=2$,		Increased reaction at bearing *A*	304	176	103	76	62	52.7
(b_2)		Increase of maximum stress.	Aft L. P. main journal	968	858	759	709	676	653
(c_2)			Aft L. P. web	2,421*	1,893	1,465	1,252	1,124	1,029
(d_2)			L. P. crank pin	1,832† / 4,048‡	2,027 / 3,161	1,880 / 2,392	1,700 / 2,008	1,543 / 1,775	1,425 / 1,594
(e_2)			Forward L. P. web	2,561*					

	Increase of maximum stress.						
(e_2)		Forward L. P. web	7,561*	5,902	4,517	3,767	3,355	3,114
(f_2)		Forward L. P. main journal	6,391	5,137	4,133	3,662	3,389	3,173
	$n=8.$							
	Maximum stress when all bearings are in line.						
(b'_2)		Aft L. P. main journal	15,000	15,000	15,000	15,000	15,000	15,000
(c'_2)		Aft L. P. web	13,636*	13,636	13,636	13,636	13,636	13,636
(d'_2)		L. P. crank pin	15,000	15,000	15,000	15,000	15,000	15,000
(e'_2)		Forward L. P. web	11,693*	11,894	12,105	12,248	12,350	12,448
(f'_2)		Forward L. P. main journal	13,074	13,041	12,930	12,846	12,749	12,668
(a_2)		Increased reaction at bearing *A*.	234	137	80	60	50	43
	Increase of maximum stress.						
(b_2)		Aft L. P. main journal	924	822	728	683	656	639
(c_2)		Aft L. P. web	2,176*	1,693	1,297	1,117	1,009	928
(d_2)		L. P. crank pin	1,937† 3,701‡	1,991 2,849	1,744 2,116	1,548 1,769	1,405 1,557	1,280 1,408
(e_2)		Forward L. P. web	6,813*	5,273	3,989	3,342	3,021	2,820
(f_2)		Forward L. P. main journal	5,777	4,674	3,784	3,384	3,147	2,988
	$n=10.$							
	Maximum stress when all bearings are in line.						
(b'_2)		Aft L. P. main journal	15,000	15,000	15,000	15,000	15,000	15,000
(c'_2)		Aft L. P. web	13,636*	13,636	13,636	13,636	13,636	13,636
(d'_2)		L. P. crank pin	15,000	15,000	15,000	15,000	15,000	15,000
(e'_2)		Forward L. P. web	11,567*	11,764	11,797	12,165	12,291	12,350
(f'_2)		Forward L. P. main journal	13,252	13,182	13,086	12,992	12,867	12,815

The stresses in the metal are for the 1st order of cranks.
All the figures are pounds per square inch, and are constant for similar engines with the same steam pressure and piston speed.

(a_1)	Increased reaction at bearing A	509	305	177	129	103	88
(b_1)	Increase of maximum stress. — Aft L. P. main journal	1,033	949	842	783	741	709
(c_1)	Aft L. P. web	2,964*	2,402	1,873	1,607	1,439	1,311
(d_1)	L. P. crank pin	1,515† / 4,677‡	1,979 / 3,864	2,064 / 3,003	1,979 / 2,576	1,872 / 2,298	1,758 / 2,081
(e_1)	Forward L. P. web	9,147*	7,445	5,794	4,913	4,335	4,057
(f_1)	Forward L. P. main journal	7,600	6,287	5,061	4,452	4,051	3,765
	$\mathfrak{D}=5.$						
(b'_1)	Maximum stress when all bearings are in line — Aft L. P. main journal	15,000	15,000	15,000	15,000	15,000	15,000
(c'_1)	Aft L. P. web	13,636*	13,636	13,636	13,636	13,636	13,636
(d'_1)	L. P. crank pin	15,000	15,000	15,000	15,000	15,000	15,000
(e'_1)	Forward L. P. web	12,035*	12,155	12,350	12,444	12,560	12,920
(f'_1)	Forward L. P. main journal	12,736	12,708	12,632	12,543	12,458	12,400
(a_2)	Increased reaction at bearing A	304	176	103	76	62	52.7
(b_2)	Increase of maximum stress. — Aft L. P. main journal	968	858	759	709	676	653
(c_2)	Aft L. P. web	2,421*	1,893	1,465	1,252	1,124	1,029
(d_2)	L. P. crank pin	1,832† / 4,048‡	2,027 / 3,161	1,880 / 2,392	1,700 / 2,008	1,543 / 1,775	1,425 / 1,594
(e_2)	Forward L. P. web	7,561*	5,902	4,517	3,767	3,355	3,114
(f_2)	Forward L. P. main journal	6,391	5,137	4,133	3,662	3,389	3,173
	$\mathfrak{D}=8.$						

(b'_2)	Maximum stress when all bearings are in line.	Aft L. P. main journal	15,000	15,000	15,000	15,000	15,000	15,000
(c'_2)		Aft L. P. web	13,636*	13,636	13,636	13,636	13,636	13,636
(d'_2)		L. P. crank pin	15,000	15,000	15,000	15,000	15,000	15,000
(e'_2)		Forward L. P. web	11,693*	11,894	12,105	12,248	12,350	12,448
(f'_2)		Forward L. P. main journal	13,074	13,041	12,930	12,846	12,749	12,668
(a_2)	Increase of maximum stress.	Increased reaction at bearing A.	234	137	80	60	50	43
(b_2)		Aft L. P. main journal	924	822	728	683	656	639
(c_2)		Aft L. P. web	2,176*	1,693	1,297	1,117	1,009	928
(d_2)		L. P. crank pin	1,937† 3,701‡	1,991 2,849	1,744 2,116	1,548 1,769	1,405 1,557	1,280 1,408
(e_2)		Forward L. P. web	6,813*	5,273	3,989	3,342	3,021	2,820
(f_2)		Forward L. P. main journal	5,777	4,674	3,784	3,384	3,147	2,988
(b'_2)	Maximum stress when all bearings are in line.	Aft L. P. main journal	15,000	15,000	15,000	15,000	15,000	15,000
(c'_2)		Aft L. P. web	13,636*	13,636	13,636	13,636	13,636	13,636
(d'_2)		L. P. crank pin	15,000	15,000	15,000	15,000	15,000	15,000
(e'_2)		Forward L. P. web	11,567*	11,764	11,797	12,165	12,291	12,350
(f'_2)		Forward L. P. main journal	13,252	13,182	13,086	12,992	12,867	12,815

$\mathfrak{B}$=10.

The stresses in the metal are for the 1st order of cranks.

All the figures are pounds per square inch, and are constant for similar engines with the same steam pressure and piston speed.

The values of the rise of stress in the round parts of the shaft may be considered as mean values over the range of variation which usually occurs in practice, as explained in foot-note p. 708, Vol. VIII.

Position of L. P. crank. For the forward main journal, lines (f) and (f'), it is at about 27° past bottom center; and for all other parts at about 27° before top center.

* In lines (c), (c'), (e) and (e'$_2$) there is no allowance for increase of stress due to change of section. $15{,}000 \div 1.1 = 13{,}636$. In lines (e), the reaction at bearing A due to want of alignment is supposed parallel to crank.

† In this line the reaction at bearing A, due to want of alignment, is supposed vertical.

‡ In this line the reaction at bearing A, due to want of alignment, is supposed parallel to crank.

TABLE XX.—STRESSES IN SHAFTS UNDER EXTREME CONDITIONS OF WANT OF ALIGNMENT OF BEARINGS

TABLE XX.—STRESSES IN SHAFTS UNDER EXTREME CONDITIONS OF WANT OF ALIGNMENT OF BEARINGS.

Line.		$F_{lb.}$	10	20	40	60	80	100
(a_1)	$\mathfrak{D}$=5.	Maximum stress in L. P. crank pin	15,894† 17,760‡	16,940 18,787	18,488 20,075	19,611 21,002	20,448 21,687	21,135 22,262
(b_1)		Maximum stress in forward L. P. web	17,432‡	19,453	22,141	23,892	25,175	27,079
(c_1)		Maximum stress in forward L. P. main journal	17,220	18,870	21,184	22,916	24,245	25,540
(d_1)		Corresponding error of bearing *B*	.59*	.98	1.69	2.33	2.91	3.49
(a_2)	$\mathfrak{D}$=8.	Maximum stress in L. P. crank pin	16,813† 19,008‡	18,446 20,373	20,471 21,961	21,715 22,932	22,468 23,590	23,108 24,070
(b_2)		Maximum stress in forward L. P. web	19,178‡	21,929	25,249	27,129	28,587	30,168
(c_2)		Maximum stress in forward L. P main journal	19,401	21,774	24,957	27,312	29,151	30,724
(d_2)		Corresponding error of bearing *B*	.99*	1.70	2.91	3.95	4.84	5.69
(a_3)	$\mathfrak{D}$=10.	Maximum stress in L. P. crank pin	17,479† 19,737‡	19,360 21,239	21,540 22,936	22,740 23,845	23,430 24,342	23,934 24,827
(b_3)		Maximum stress in forward L. P. web	20,287‡	23,311	26,755	28,875	30,419	32,030
(c_3)		Maximum stress in forward L. P. main journal	20,646	23,418	27,276	29,912	31,749	33,671
(d_3)		Corresponding error of bearing *B*	1.28*	2.19	3.75	5.00	6.00	6.98

This table is deduced from Table XIX by supposing an error of bearing *B* which would raise the maximum reaction at bearing *A* to 700 pounds per square inch. Stresses are augmented and in pounds per square inch.

† In this line the reaction at bearing *A* due to want of alignment is supposed vertical.

‡ In this line the reaction at bearing *A* due to want of alignment is supposed parallel to crank.

* The error of bearing *B* is in thousandths of an inch per one inch stroke of engine.

TABLE XXI.—MOMENTS DUE TO STEAM, WITH BEARING *B* REMOVED.

Line.		Column	I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.
(a)		Part of crank shaft	Aft L.P. main journal.		Aft L.P. crank web.		L.P. crank pin.		Forward L.P. crank web.			Forward L.P. main journal.		
(b)		Moment	Turning.	Bending.	Turning.	Bending.	Turning.	Bending.	Turning.	Bending.		Turning.	Bending.	
(c)		Point of revolution at which moment is measured	L.P. crank 27° before top center.											
(d)		Value of moment from Figs. III, XXIII, and XXIV	$.738F_m$	$.484F_mk$	$.725F_m$	$.484\times.9F_mk$	$.606F_m$	$.484\times.45F_mb$	$.606F_m$	$.484\times.9F_mK-.478\times.9F_mK_1$		$.606F_m$	$.484F_mH-.478F_mH_1$	
(e)	Multipliers.	Ten per cent. loss from boiler pressure to absolute zero	.9	.9	.9	.9	.9	.9	.9	.9	.9	.9	.9	.9
(f)	Multipliers.	Increase for unequal steam stresses	1.06	1.1	1.06	1.1	1.06	.....	1.06	.....	.....	1.06	.....	.....
(h)	Multipliers.	Shearing stress augmented in ratio 7 to 10	10/7	.....	.....	.....	10/7	.....	.....	.....	.....	10/7	.....	.....
(k)	Multipliers.	Increase for change of section	1.05	1.1	.....	.....	.....	.....	.....	.....	.....	1.05	1.1	1.1
(l)		Value of moment taken in calculation	$1.056F_m$	$.527F_mb$	$.692F_m$	$.431F_mk$	$.826F_m$	$.196F_mb$	$.578F_m$	$(.392K-.387K_1)F_m$		$.867F_m$	$(.479H-.473H_1)F_m$	

Bearings *A*, *C* and *D* are supposed to be in line.

This table is for the 2d order of cranks.

TABLE XXII.—MOMENTS DUE TO INERTIA WITH BEARING *B* REMOVED.

Line.	Column		I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.
(a)	Part of crank shaft		Aft L.P. main journal.		Aft L.P. crank web.		L.P. crank pin.		Forward L.P. crank web.			Forward L.P. main journal.		
(b)	Moment		Turning.	Bending.	Turning.	Bending.	Turning.	Bending.	Turning.	Bending.		Turning.	Bending.	
(c)	Point of revolution at which moment is measured		L. P. crank 27° before top center.		L.P. crank 22½° before top center.									
(d)	Value of moment from Figs. X, XI, XIII and XXV		$.261G$	$.578Gk$	$.219G$	$.615\times.9Gk$	$.0804G$	$.615\times.45Gb$	$.0705G$	$.615\times.9GK-1.052\times.9GK_1$		$-.0282G$	$.615GH-1.052GH_1$	
(e)	Multipliers.	Ratio of parts producing turning, to those producing bending moment	.72	.....	.72	.....	.72	.....	.72	.....	.....	.72	.....	.....
(f)		Increase for excess of speed	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1
(h)		Shearing stress augmented in ratio 7 to 10	10/7	.....	.....	.....	10/7	.....	.....	.....	.....	10/7	.....	.....
(k)		Increase for change of section	1.05	1.1	.....	.....	.....	.....	.....	.....	.....	1.05	1.1	1.1
(l)		Increase for moving masses being heavier than estimated	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05
(m)	Value of moment taken in calculation		$.326G$	$.734Gk$	$.182G$	$.639Gk$	$.0955G$	$.320Gb$	$.0586G$	$(.639K-1.094K_1)G$		$-.0352G$	$(.781H-1.337H_1)G$	

Bearings *A*, *C* and *D* are supposed to be in line. This table is for the 2d order of cranks.

TABLE XXIII.—FORCES F_1, F_2 AND F_3 FOR CALCULATING DEFLECTION AT B WHEN THIS BEARING IS REMOVED.

Line.	Column		I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.
(a)	Order of cranks		1st order.						2d order.					
(b)	Force		F_1		F_2		F_3		F_1		F_2		F_3	
(c)	Steam or inertia		Steam.	Inertia.	Steam.	Inertia.	Steam.	Inertia.	Steam.	Inertia.	Steam.	Inertia.	Steam.	Inertia.
(d)	Point of revolution at which force is measured		L. P. crank before top center.						L. P. crank before top center.					
			27°	22½°	27°	22½°	27°	22½°	27°	22½°	27°	22½°	27°	22½°
(e)	Values from Figs. X and XXIV		$-.989F_m$	$+.553G$	$+.767F_m$	$+.256G$	$-.478F_m$	$-1.052G$	$+.767F_m$	$+.211G$	$-.989F_m$	$+.640G$	$-.478F_m$	$-1.052G$
(f)	Multipliers.	Ten per cent. loss from boiler pressure to absolute zero	.9	.....	.9	.....	.9	.....	.9	.....	.9	.....	.9	.....
(g)		Changes due to unequal steam stresses	1.0	.....	.9	.....	1.1	.....	.9	.....	1.0	.....	1.1	.....
(h)		Increase for excess of speed	.....	1.1	.....	1.1	.....	1.1	.....	1.1	.....	1.1	.....	1.1
(k)		Increase for moving masses being heavier than estimated	.....	1.05	.....	1.05	.....	1.05	.....	1.05	.....	1.05	.....	1.05
(l)	Value of force taken in calculation		$-.890F_m + 639G$		$+.621F_m + .296G$		$-.473F_m - 1.215G$		$+.621F_m + .244G$		$-.890F_m + .739G$		$-.473F_m - 1.215\ G$	

Bearings A, C and D are supposed to be in line.

TABLE XXV.—STRESSES IN SHAFTS, FIGS. XIV TO XXI, WHEN BEARING *B* IS REMOVED.

Line	F m.		10	20	40	60	80	100
(a_1)	F m.=2.	Maximum stress in aft L. P. journal	.....	.....	15,062	15,163	15,272	.....
(b_1)		Maximum stress in aft L. P. web	.....	.....	.....	.....	.....	.....
(c_1)		Maximum stress in L. P. crank pin	.....	.....	.....	15,137	15,384	15,570
(d_1)		Maximum stress in forward L. P. web	.....	.....	.....	.....	15,429	15,914
(e_1)		Maximum stress in forward L. P main journal	.....	.....	.....	.....	15,132	15,730
(f_1)		Maximum deflection at bearing *B*.. 2d order of cranks	.278	.437	.719	.978	1 220	1.454
(g_1)		Maximum deflection at bearing *B*.. 1st order of cranks	.044	.069	.114	.155	.194	.231
(a_2)	F m.=5.	Maximum stress in aft L. P. journal	.....	.....	.....	15,179	.....	.....
(b_2)		Maximum stress in aft L. P. web	.....	.....	.....	.....	.....	.....
(c_2)		Maximum stress in L. P. crank pin..	.....	.....	15,058	15,486	15,812	16,058
(d_2)		Maximum stress in forward L. P. web	.....	.....	.....	15,410	16,122	16,646
(e_2)		Maximum stress in forward L. P. main journal	.....	.....	.....	.....	16,040	17,002
(f_2)		Maximum deflection at bearing *B*.. 2d order of cranks	.463	.772	1.331	1.829	2.292	2.732
(g_2)		Maximum deflection at bearing *B*.. 1st order of cranks	.216	.360	.620	. 852	1.068	1.273
(a_3)	F m.=8.	Maximum stress in aft L. P. journal	.....	.....	15,038	.....	.....	.....

(g_2)	F_m=8.	Maximum deflection at bearing B — 1st order of cranks	.216	.360	.620	.852	1.068	1.273
(a_3)	F_m=8.	Maximum stress in aft L. P. journal	.....	.....	15,038	.....	.....	.....
(b_3)	F_m=8.	Maximum stress in aft L. P. web	.....	.....	.....	.....	.....	.....
(c_3)	F_m=8.	Maximum stress in L. P. crank pin	.....	.....	15,432	15,898	16,225	16,452
(d_3)	F_m=8.	Maximum stress in forward L. P. web	.....	.....	.....	15,951	16,631	17,198
(e_3)	F_m=8.	Maximum stress in forward L. P. main journal	.....	.....	.....	16,112	17,491	18,600
(f_3)	F_m=8.	Maximum deflection at bearing B — 2d order of cranks	.663	1.144	1.959	2.656	3.257	3.821
(g_3)	F_m=8.	Maximum deflection at bearing B — 1st order of cranks	.443	.764	1.308	1.774	2.176	2.552
(a_4)	F_m=10.	Maximum stress in aft L. P. journal	.....	.....	15,105	.....	.....	.....
(b_4)	F_m=10.	Maximum stress in aft L. P. web	.....	.....	.....	.....	.....	.....
(c_4)	F_m=10.	Maximum stress in L. P. crank pin	.....	.....	15,663	16,132	16,418	16,620
(d_4)	F_m=10.	Maximum stress in forward L. P. web	.....	.....	15,210	16,215	16,929	17,436
(e_4)	F_m=10.	Maximum stress in forward L. P. main journal	.....	.....	15,046	16,937	18,361	19,612
(f_4)	F_m=10.	Maximum deflection at bearing B — 2d order of cranks	.805	1.381	2.343	3.131	3.792	4.380
(g_4)	F_m=10.	Maximum deflection at bearing B — 1st order of cranks	.618	1.060	1.799	2.405	2.912	3.364

Bearings A, C and D are supposed to be in line.

The stresses in this table are for the 2d order of cranks. The L. P. crank, for both 1st and 2d orders, is about 27° before top center. The stresses are augmented and are in pounds per square inch. They are constant for similar engines with the same steam pressure and piston speed. No stresses are noted which are under 15,000 pounds per square inch; and none are noted in lines (a) and (b) unless they are larger than the stresses, for the same value of F_m, in lines (e) and (d) respectively. These conditions exclude all values from lines (b).

The deflections in lines (f) and (g) are in thousandths of an inch per 1 inch stroke of engine.

As the stresses for the 1st order of cranks is not much less than for the 2d, when the L. P. is about 27° before top center, all cases are covered by taking the stresses in lines (c), d) and (e) along with the deflections in line (g).

TABLE XXIV.—TOTAL PRESSURE ON BEARING *A* WITH BEARING *B* REMOVED.

Line	Column		I.	II.
(a)	Steam or inertia		Steam.	Inertia.
(b)	Point of revolution at which pressure is measured		L. P. crank before top center. 27°	18°
(c)	Value of pressure from Figs. XXIV and XXV		$.484F_m$	$.650G$
(d)	Multipliers.	Ten per cent. loss of pressure from boiler pressure to absolute zero	.9	.....
(e)		Increase for unequal steam stresses	1.1	.....
(f)		Increase for excess of speed	.....	1.1
(g)		Increase for moving masses being heavier than estimated	.....	1.05
(h)	Value of total pressure taken in calculation		$.479F_m + .751G$	

Bearings *A*, *C* and *D* are supposed to be in line.
This table is for the 2d order of cranks.

TABLE XXVI.—PRESSURE ON BEARING *A* WITH BEARING *B* REMOVED.

Line	[illegible]	2	5	8	10
(a)	Nominal pressure	575	539	524	518
(b)	Nominal pressure divided by 400	1.44	1.35	1.31	1.29

Bearings *A*, *C* and *D* are supposed to be in line.
Nominal pressures are in pounds per square inch.
The figures in this table are constant for similar engines of the same steam pressure and piston speed.
This table is for the 2d order of cranks, and the L. P. crank about 27° before top center. The pressures for the 1st order of cranks are very slightly higher than these.

TABLE XXIX.—SHEARING STRESS IN ISOTROPIC RECTANGULAR PRISM.

Short side = one inch. Twisting moment = one inch pound.

Long side in inches = $2a$		1.5	1.6	1.7	1.8	1.9	2.0	2 1	2.2	2.3
Shearing stress on long side = T.	$x=0$ $y=.5$	2.886	2.667	2.478	2.310	2.164	2.034	1.917	1.812	1.718
	$x=.2a$ $y=.5$	2.827	.....	.....	.....	2.129	.....	.....	.....	1.696
	$x=.4a$ $y=.5$	2.634	.....	2.280	.....	2.009	.....	1.794	.....	1.620
	$x=.6a$ $y=.5$	2.256	.....	.....	.....	1.757	.....	.....	.....	1.446
	$x=.8a$ $y=.5$	1.570	.....	1.394	.....	1.259	.....	1.153	.....	1.067
	$x=.9a$ $y=.5$	1.014	.....	.....	.....	.826	.....	.....	.....	.714
	$x=.95a$ $y=.5$	.629	.....	.....	.....	.512	.....	.....	.....	.450
Shearing stress on short side = S.	$x=a$ $y=0$	2.479	.....	2.049	.....	1.741	.....	1.509	.....	1 330
	$x=a$ $y=.1$	2.413	.....	1.994	.....	1.694	.....	1.468	.....	1.294
	$x=a$ $y=.2$	2.206	.....	1.822	.....	1.547	.....	1.341	.....	1.182
	$x=a$ $y=.3$	1.832	.....	1.511	.....	1.282	.....	1.111	.....	.980
	$x=a$ $y=.4$	1.219	.....	1.004	.....	.851	.....	.737	.....	.650
	$x=a$ $y=.45$	.760	.....	.625	.....	.530	.....	.459	.....	.404
Diameter of right circular cylinder of equal strength, in inches		1.208	1.241	1.271	1.301	1.330	1.358	1.385	1.411	1.437
$M/\mu\tau$		.2936	.3260	.3586	.3914	.4243	.4574	.4905	.5237	.5569

S and T are in pounds per square inch. They vary directly as M and inversely as the cube of the linear dimensions of the section.
$M/\mu\tau$ is independent of the unit of force. It varies directly as the fourth power of the linear dimensions.

APPENDIX A.

ON THE TORSION OF AN ISOTROPIC RECTANGULAR PRISM.

The curves and table in this appendix are given chiefly for their intrinsic value.

As explained early in the paper, the shearing stress arising from torsion was found to have no influence on the maximum stress in the webs, as that maximum occurred at the edges where the shearing stress is zero. It has a slight but negligible influence on the shearing stress in the main journal. In other forms of crank shaft the shearing stress in the webs might be important.

The following solution of this problem is due to St. Venant.*—

In Fig. XXVI, $ABDC$ is a section of a rectangular prism normal to its longitudinal axis through the center O. The long side $AB = 2a$, and the short side $AC = 2b$.

The axis of z is taken along the axis of the prism, and the axes Ox and Oy are parallel to AB and CA respectively.

Let M be the moment of torsion of the prism; that is, the moment applied oppositely to each end, twisting the prism about the axis of z.

Let τ be the angle of twist in circular measure per unit length of prism.

μ denotes the rigidity modulus of the material. It may be taken as 12×10^6 pounds per square inch for steel, though its numerical value does not affect the results in this appendix. Let P, Q and R be tensions at the point (x, y, z), per unit area, normal to the planes $x = Const.$, $y = Const.$ and $z = Const.$

Let S be the shearing stress normal to Ox across the planes $y = Const.$ and $z = Const.$ The equality of the shearing stresses normal to Ox across these two planes is readily proved—see Vol. VIII, page 673 and Fig. I.

S and τ are in pounds per square inch. They vary directly as M and inversely as the cube of the linear dimensions of the section. $M/\mu\nu$ is independent of the unit of force. It varies directly as the fourth power of the linear dimensions.

* See Mr. Love's Treatise referred to on page 682, Vol. VIII. I shall refer frequently to Mr. Love's Treatise in this, and subsequent appendices; and, for convenience of reference, the numbers of his equations and articles are given in square brackets—thus, [VI, 69] means Chapter VI, equation (69); and [§ 43] means article 43.

Let T be the shearing stress normal to Oy across the planes $z = Const.$ and $x = Const.$

Let U be the shearing stress normal to Oz across the planes $x = Const.$ and $y = Const.$

St. Venant's solution involves the conditions,

$$P = Q = U = 0; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [VI, 3]$$

that is, that there are no normal tractions across longitudinal planes. (If U were not zero there would be normal tractions across the longitudinal planes $x \pm y = 0$.) This assumes, either that the ends are so distant that the distribution of stress on them has no influence on its distribution in the part of the prism considered, or that the end stresses are distributed in the manner given below by equations (32) and (33). The influence of end distribution would rapidly die out as we recede from them.

$$M = \frac{16}{3}\mu\tau\, ab^3 - \mu\tau\, b^4 \left(\frac{4}{\pi}\right)^5 \sum_{n=0}^{n=\infty} \frac{tanh\, \frac{(2n+1)\pi a}{2b}}{(2n+1)^5}; \quad . \quad [VI, 69]$$

$$\left.\begin{aligned} \text{and } S &= \mu\left(\frac{\partial \varphi}{\partial y} + \tau x\right) \\ T &= \mu\left(\frac{\partial \varphi}{\partial x} - \tau y\right) \end{aligned}\right\}, \quad . \quad . \quad \left[\begin{matrix} \text{VI, 25; with } \beta_0 = -\tau, \\ \beta_1 = \beta_2 = 0 \text{ and } L = M = \mu. \end{matrix}\right]$$

where $\varphi = -\tau xy$

$$+ 4\tau b^2 \left(\frac{2}{\pi}\right)^3 \sum_{n=0}^{n=\infty} \left\{ \frac{(-1)^n}{(2n+1)^3} \frac{sinh\, \frac{(2n+1)\pi x}{2b}}{cosh\, \frac{(2n+1)\pi a}{2b}} sin\, \frac{(2n+1)\pi y}{2b} \right\} \quad . \quad . \quad . \quad [VI, 66]$$

Performing the differentiations and writing out the series we get,

$$M / \mu\tau = \frac{16}{3} ab^3 - 3.3462\, b^4 \left\{ \frac{e^{\frac{\pi a}{2b}} - e^{-\frac{\pi a}{2b}}}{e^{\frac{\pi a}{2b}} + e^{-\frac{\pi a}{2b}}} \right.$$

$$\left. + \frac{e^{\frac{3\pi a}{2b}} - e^{-\frac{3\pi a}{2b}}}{3^5 \left(e^{\frac{3\pi a}{2b}} + e^{-\frac{3\pi a}{2b}}\right)} + \frac{e^{\frac{5\pi a}{2b}} - e^{-\frac{5\pi a}{2b}}}{5^5 \left(e^{\frac{5\pi a}{2b}} + e^{-\frac{5\pi a}{2b}}\right)} + \&c. \right\} \; . \; (31)$$

$$S/\mu\tau = 1.6211\, b \left\{ \frac{e^{\frac{\pi x}{2b}} - e^{-\frac{\pi x}{2b}}}{e^{\frac{\pi a}{2b}} + e^{-\frac{\pi a}{2b}}} \cos\frac{\pi y}{2b} \right.$$

$$\left. - \frac{e^{\frac{3\pi x}{2b}} - e^{-\frac{3\pi x}{2b}}}{3^2\left(e^{\frac{3\pi a}{2b}} + e^{-\frac{3\pi a}{2b}}\right)} \cos\frac{3\pi y}{2b} + \frac{e^{\frac{5\pi x}{2b}} - e^{-\frac{5\pi x}{2b}}}{5^2\left(e^{\frac{5\pi a}{2b}} + e^{-\frac{5\pi a}{2b}}\right)} \cos\frac{5\pi y}{2b} - \&c. \right\} \quad . . (32)$$

$$T/\mu\tau = -2y + 1.6211\, b \left\{ \frac{e^{\frac{\pi x}{2b}} + e^{-\frac{\pi x}{2b}}}{e^{\frac{\pi a}{2b}} + e^{-\frac{\pi a}{2b}}} \sin\frac{\pi y}{2b} \right.$$

$$\left. - \frac{e^{\frac{3\pi x}{2b}} + e^{-\frac{3\pi x}{2b}}}{3^2\left(e^{\frac{3\pi a}{2b}} + e^{-\frac{3\pi a}{2b}}\right)} \sin\frac{3\pi y}{2b} + \frac{e^{\frac{5\pi x}{2b}} + e^{-\frac{5\pi x}{2b}}}{5^2\left(e^{\frac{5\pi a}{2b}} + e^{-\frac{5\pi a}{2b}}\right)} \sin\frac{5\pi y}{2b} - \&c. \right\} \quad . . (33)$$

The curves in Figs. XXVI and XXVII have been deduced from equations (31), (32) and (33); sufficient points being calculated and the rest got by interpolation.

In Fig. XXVI, $AB = 1.5 \times AC$. For this case the distribution of shearing stress over the whole section has been determined. In the quadrant $OEBH$, lines marked 1, 2, &c., are drawn parallel to Ox at $b/10$ apart and form the bases for the curves of shearing stress correspondingly numbered. Thus at the point P, in the line marked 7, the total shearing stress (got by combining the shearing stresses S and T at the point) is given by the ordinate $\mathbf{S}$. Lines are drawn parallel to Oy at distances .2, .4, .6, .8, .9, .95 of a from Oy. At the intersection of these two sets of lines the arrows indicate the direction of the shear.

The end curve, drawn on HB as base, gives the shearing stress along the short end. The shearing stress at any boundary must always be tangential.

In the quadrant $OEAG$ are given the curves of equal shearing stress. Considered as contour lines, it will be seen that they map out an anticlastic surface.

The scale gives unaugmented shearing stresses in pounds per square inch per one inch pound twisting moment when the short side AC = one inch. These stresses are, of course, proportional to the twisting moment, and inversely proportional to the cube of the linear dimensions of the section.

The shearing stress is seen to be a maximum at E and there is a smaller maximum at H. At the angular points A, B, C, D, it is zero; and this is always the case at the projecting angle of any cylinder under torsion.

The circle shown in this figure is the section of the cylinder of equal *strength*.

Fig. XXVII gives the curves of shearing stress at the boundaries for values of the ratio $a/b = 1.5, 1.6, \ldots, 2.3$, marked at the ends of the curves. They are all shown on the same base EB for the long side, and on BH for the short side. B is an angle of the prism. E and H are the centers of the long and short sides, respectively. The shearing stresses at points P, for $a/b = 1.9$, are given by ordinates $\mathbf{S}$. The stresses are to be read from the scale in the same way as for Fig. XXVI.

The arcs of circles give the radii of cylinders of equal *strength*. The ratio of a/b to which each circle corresponds is marked at the end.

In addition to the curves the results calculated are given in Table XXIX, for their more accurate preservation.

After calculating nine terms of the series in equation (32) when $x = a$, it was found that the convergence was not sufficient to ascertain with certainty the sums of the whole series to the third decimal place without taking a very large number of terms. The necessary correction for the sum of the nine terms was found as follows:—

In all cases calculated $e^{\frac{3\pi a}{2b}}$ is very large compared with $e^{-\frac{3\pi a}{2b}}$; and, well within the limits of accuracy of Table XXIX, equation (32) reduced to

$$S/\mu\tau =$$

$$1.6211\, b \left\{ \frac{e^{\frac{\pi a}{2b}} - e^{-\frac{\pi a}{2b}}}{e^{\frac{\pi a}{2b}} + e^{-\frac{\pi a}{2b}}} \cos\frac{\pi y}{2b} - \frac{1}{3^2}\cos\frac{3\pi y}{2b} + \frac{1}{5^2}\cos\frac{5\pi y}{2b} - \&c. \right\}.$$

Thus the sum of all terms after the first is independent of the ratio a/b. It was only necessary, then, to calculate for $x = a$ and $a/b = 1.5$ from the following more convergent series,

$$S/\mu\tau = +2x - 1.6211\ a\left\{\frac{e^{\frac{\pi y}{2a}} + e^{-\frac{\pi y}{2a}}}{e^{\frac{\pi b}{2a}} + e^{-\frac{\pi b}{2a}}} \sin\frac{\pi x}{2a}\right.$$

$$\left. - \frac{e^{\frac{3\pi y}{2a}} + e^{-\frac{3\pi y}{2a}}}{3^2\left(e^{\frac{3\pi b}{2a}} + e^{-\frac{3\pi b}{2a}}\right)} \sin\frac{3\pi x}{2a} + \frac{e^{\frac{5\pi y}{2a}} + e^{-\frac{5\pi y}{2a}}}{5^2\left(e^{\frac{5\pi b}{2a}} + e^{-\frac{5\pi b}{2a}}\right)} \sin\frac{5\pi x}{2a} - \&c.\right\}.$$

This equation is got from equation (33) by interchanging a with b and x with y, and also changing the signs of the terms to keep S of the same sign as in equation (32).

The correct values for $a/b = 1.5$, $x = a$ and $y = 0$, $.1\,b$, &c., readily furnished the slight corrections for all the other values of the ratio a/b.

The stress at points in the interior of the prism, Fig. XXVI, were got by graphically reducing the exponential factors of each term—such as

$$\frac{e^{\frac{\pi x}{2b}} - e^{-\frac{\pi x}{2b}}}{e^{\frac{\pi a}{2b}} + e^{-\frac{\pi a}{2b}}}$$

—in the ratios expressed by the circular functions—such as $cos\,\frac{\pi y}{2b}$. The total values of S and T so found were then combined and gave the ordinates of the curves. Their ratios gave the directions of total shearing stress shown by the arrows. It is not considered of sufficient importance to tabulate the values of the exponential factors by which this result was obtained.

(*To be continued.*)

THE CONTRACT TRIALS OF THE U. S. GUNBOATS *HELENA* AND *WILMINGTON.*

By Passed Assistant Engineer W. S. Smith, U. S. N.

The steel, twin screw gunboats *Helena* and *Wilmington* were built by the Newport News Shipbuilding and Dry Dock Company at their works in Newport News, Virginia. The *Helena* was launched on January 30, 1896, and the *Wilmington*, on October 19, 1895.

The designs and specifications were furnished by the Navy Department, the contract price for each vessel being $280,000, complete in every respect, except armament.

They are sister ships built from the same model and are alike in every dimension. They have high freeboard forward, given by an upper deck extending aft about three-fourths of the vessel's length, providing increased berthing space and protection to the battery. This deck is cut away at frame 88, and the remaining portion of the main deck is that part immediately over the cabin and wardroom. There is a double bottom under the machinery space, but it does not extend outboard under the fore-and-aft bunkers.

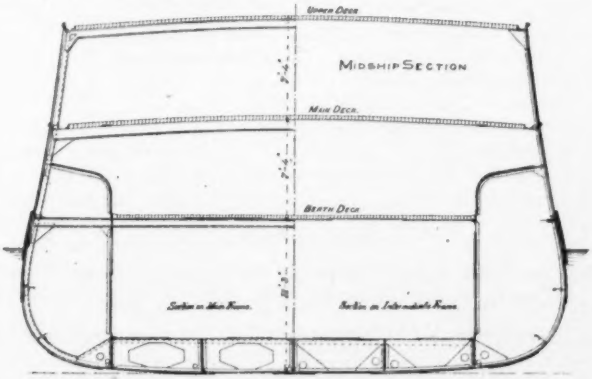
The bunkers are arranged to give coal protection to the engines and boilers, and extend continuously above the berth deck and finish just below the line of air ports under the main deck. There is a trimming tank in the bow and one in the stern, and one fresh water tank on each side of the engine space. These vessels are designed for service in rivers and estuaries and are provided with a large military mast, which is fitted with a spiral stairway. Besides the conning tower, which is a part of this mast and well elevated, there are two fighting platforms, and a third platform between the two for the search light. The machine and rapid fire guns situated in these fighting tops are intended to command the high banks of a river and the adjacent fields.

There are seven boats: a steam cutter and sailing launch, carried on deck abreast of the smoke pipe, one on each side; two sailing cutters, carried on the upper deck abreast of their davits and just forward of the break of the deck; two whale boats, carried on davits from the main deck aft; and one dinghy, carried in the sailing launch.

The main battery consists of eight 4-inch rapid fire rifles; two on the forecastle, four in sponsons on the main deck underneath the upper deck, and two on the main deck aft. Those in the open on the forecastle and main deck aft have shields attached to their carriages, while those on the main deck underneath are protected by sponson armor of 2½-inch nickel steel plate.

The secondary battery is composed of four 6-pounder rapid fire guns, four 1-pounders and four Gatling guns. Two of the 6-pounders are placed on the main deck between the 4-inch R. F. rifles behind one-inch nickel steel shields, and the other two, on the upper deck abreast of the chart house. The 1-pounders are placed on the first fighting platform above the conning tower, and the Gatling guns in the higher fighting top. The guns on the main deck under cover have a range of seventy degrees both forward and abaft the beam, those on the forecastle have a range from ahead to seventy degrees abaft the beam, while the 4-inch guns on the quarter deck aft have a corresponding range from direct aft.

There are no torpedo tubes.



In addition to the coal protection, there is worked a strake of one-inch nickel steel plating at the water line and outside of the hull plating. This belt is 3 feet 8 inches wide and extends over the space occupied by the engines and boilers.

The contract required an average speed of 13 knots for four consecutive hours, subject to a penalty at the rate of $20,000 per knot, with a corresponding premium for any speed above that called for and guaranteed.

HULL.

Length between perpendiculars, feet and inches	250-0
on L.W.L., feet and inches	250-9¼
over all, feet and inches	251-10⅝
Beam, feet and inches	40-1⅝
Ratio of length to beam	6.28
Depth of hold, top of floors to top of main deck beams, feet and inches	16-0¾
Draught, mean, seagoing trim (normal) feet	9-0
Trim	0-10⅞
Displacement (normal), tons	1,396.65
Tons per inch, L.W.L.	17.10
Area of immersed midship section, L.W.L., square feet	340.
L.W.L. plane, square feet	7,183.
Center of gravity of L.W.L. abaft midship section, frame No. 63, feet and inches	9-11 5/16
Center of buoyancy above bottom of keel, feet and inches	5- 3 3/16
forward of midship section, frame No. 63, ft. and ins.	2- 3 1/16
gravity above bottom of keel, feet and inches	13- 1⅜
Transverse metacenter above center of buoyancy, feet and inches	14- 3⅜
Longitudinal metacenter above center of buoyancy, feet and inches	501- 0¼
Transverse G.M., metacentric height, feet and inches	6- 5 7/16
Longitudinal G.M., feet and inches	493- 2 1/16
Block coefficient for displacement	0.540
Cylindrical coefficient for displacement. (Longitudinal cylinder)	.573
Midship section coefficient	.942
L.W.L. coefficient	.726
Wetted surface to L.W.L., square feet	10,404.
Number of frames	125.
water tight compartments below main deck, and including double bottoms	48.

MAIN ENGINES.

There are twin screw engines of the vertical, inverted, direct-acting, three cylinder, triple expansion type, designed by the Navy Department. They are placed in one compartment and

arranged in the usual manner with the high pressure cylinders forward, the cranks being 120° apart and following in the order beginning with the H.P. crank. The cylinders are not steam-jacketed, but are well lagged with magnesia and wood covering. Each H.P. cylinder has one, and each I.P. cylinder, two piston valves; each L.P. cylinder has a double-ported slide valve, provided with a balance piston, the top of which connects with the condenser. The valve gear is the ordinary Stephenson link motion with double bar links.

The main engines were designed to develop 1600 I.H.P., (including the power of the main circulating pump,) when running at 283 revolutions per minute, with a steam pressure of 180 pounds per square inch in the boilers.

The framing of the engines consists of wrought steel columns tied with wrought steel ties, and supports the cylinders from a cast steel bed plate, made in three sections and bolted together. The columns support the cast iron guides for the crosshead slippers, which are of manganese bronze.

The piston rods and valve stems are oil-tempered and ground to a true cylinder. The crank shafts are in three interchangeable sections.

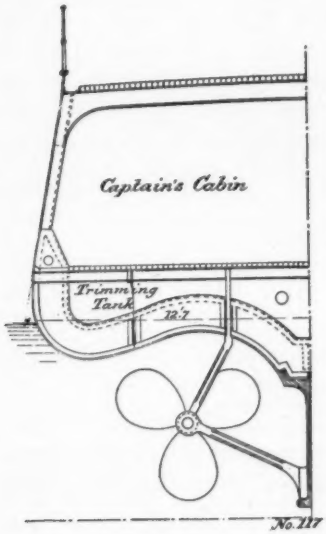
There is one condenser for both engines just abaft the low pressure cylinders and placed athwartship.

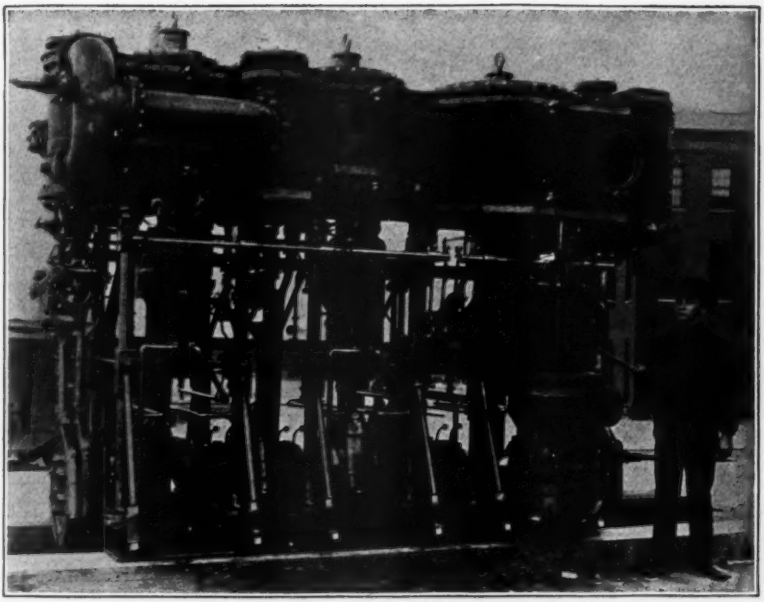
The following are some of the principal dimensions of the engines:

DATA OF ONE MAIN ENGINE (STARBOARD).

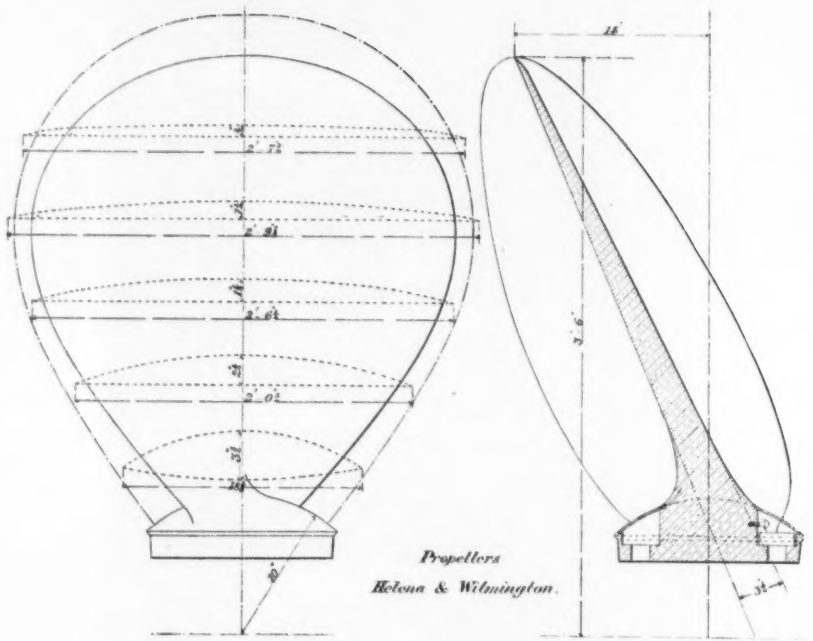
Number of cylinders		3
Diameter of cylinders, inches	H.P.	14½
	I.P.	22
	L.P.	33¾
Stroke, inches		18
Diameter of valves, inches	H.P. (one)	8½
	I.P. (two)	8½
Width and height of L.P. valve, inches		34⅛–25 7/32
Diameter of piston rods (all), inches		3
Length of connecting rods, center to center, inches		40
Distance between centers of cylinders, feet and inches		3–2
Volume swept by pistons per stroke, cubic feet, mean	H.P.	1.72
	I.P.	3.95
	L.P.	9.30

Volumetric cylinder clearance (mean of the two ends) cu. ft.,	H.P.	.41
	I.P.	.88
	L.P.	.90
Ratios of net areas of pistons	H.P. to I.P.	2.30
	I.P. to L.P.	2.35
	H.P. to L.P.	5.42
Crank shaft, length, each section, feet and inches		3–2
diameter, inches		$6\frac{1}{8}$
length, each bearing, inches		7
axial hole, diameter, inches		$3\frac{1}{4}$
Crank pins, diameter, inches		$6\frac{3}{4}$
length, inches		8
axial hole, diameter, inches		$3\frac{1}{4}$
Propeller shafts, forward section, length, feet and inches		28–$11\frac{1}{2}$
after section, length, feet and inches		35–$6\frac{1}{2}$
diameter and axial holes, forward section, inches		$6\frac{1}{8}$ and 3
after section, inches		$6\frac{1}{2}$ " $3\frac{3}{4}$
Thrust shafts, diameter, inches		$5\frac{7}{8}$
axial holes, inches		$2\frac{7}{8}$
number of thrust collars		9
thickness of thrust collars, inches		$1\frac{1}{8}$
diameter of thrust collars, inches		$8\frac{7}{8}$





TRIPLE EXPANSION ENGINE OF U. S. S. "HELENA" AND "WILMINGTON."


12'
3'-6"
Propellers
Helena & Wilmington.

PROPELLERS.

The propellers are three-bladed, the boss being made of composition and the blades of manganese bronze. The starboard propeller is right, and the port one, left-handed. The general shape of the blades is almost circular, and they are bent back on an angle, the edge of periphery being fourteen inches from the perpendicular. The run of the ship is of peculiar construction to admit of free flow of water to the screws, as shown in the cut.

Number of blades	3
Diameter, feet and inches	7–0
of hub, feet and inches	1–8
Length of hub, feet and inches	1–5
Thickness of blade at center of hub, inches	3¼
Helicoidal area, square feet, one propeller	18.7
Disc area, square feet, one propeller	38.48

Pitch as set, feet and inches:—

	Starbd.	*Port.*	*Mean.*
Helena, feet and inches	7–1	7–1½	7–1¼
Wilmington	7–2¼	7–0¾	7–1½

Ratio of pitch to diameter:—*Helena*	1.015 to 1
Wilmington	1.018 to 1

Distance of center of hub above lowest point of keel, feet and inches:—

Helena	4–4⅜
Wilmington	4–4⅝

CONDENSERS.

Main Condenser.—The main condenser shell is cylindrical, of composition, in two sections bolted together. It is placed abaft the engines and amidship, with the axis running athwartship. The circulating water passes through the tubes.

Number of tubes in condenser	2,027
Length between tube sheets, feet and inches	6–9½
Diameter of tubes, outside, inch	⅝
Cooling surface, square feet	2,246
Ratio total cooling surface to total H.S. of six main boilers	1 to 2.14

Auxiliary Condenser.—There is one auxiliary condenser of the Wheeler Admiralty type, with combined air and circulating pump. The condensing surface is 180 square feet.

Steam cylinder, diameter, inches	5½
Air pump cylinder, diameter, inches	6
Circulating pump cylinder, diameter, inches	6
Stroke, common, inches	7

Air Pump.—There is an air pump for each engine, driven by a rockshaft and arms from the low pressure crosshead. The pump cylinder is 14.75 inches in diameter, and the stroke of the piston, 6 inches. The design of this pump was shown in an article on air pumps by Passed Assistant Engineer (now Chief Engineer) F. H. Bailey, U. S. Navy, and published in the JOURNAL, page 518 of Vol. VI. As made, the diameter of the pump rod has been changed from $1\frac{7}{8}$ to $1\frac{7}{16}$ inches, and tapered in the crosshead to $1\frac{1}{8}$ instead of $1\frac{3}{8}$ inches. For the better illustration of this article the cut is here reproduced.

Centrifugal Pump.—There is one centrifugal pump, situated on the starboard side outboard of and abaft the L.P. cylinder. The diameter of the runner is 24, of the suction, 9, and of the discharge, 8 inches. It is fitted to draw from the sea, from the main drain and from the engine room bilge, and is fitted to deliver to the condenser, or, by means of a connecting pipe and by pass valve, directly overboard. The pump is driven by a single engine, with a cylinder diameter of 6 and a stroke of 5 inches. The diameter of the piston rod is $1\frac{3}{8}$ inches.

Feed and other Auxiliary Pumps.—All of the auxiliary pumps are of the Blake vertical duplex pattern, except the pump attached to the Wheeler condenser. There are in the engine room, one main feed, one fire and bilge and one sanitary pump, and in each fire room, one auxiliary feed pump. All of these pumps have steam cylinders 6 inches in diameter and a stroke of 7 inches. The water cylinders of the main feed pumps are 4 inches, and those of the other pumps, $4\frac{1}{8}$ inches in diameter.

Distilling Apparatus.—This consists of a Baird evaporator No. 3, with U-tubes, the plane of the U being horizontal, and the steam escaping by a single tube at the bottom. There are two No. $2\frac{1}{2}$, double coil, Baird distillers. A water meter is also fitted. The distiller pump is of the Blake horizontal type, with combined fresh and salt water pump cylinders. The steam cylinder

is $3\frac{1}{2}$ inches, and the water cylinder, $2\frac{1}{8}$ inches in diameter, the stroke being 4 inches. The following tests were made of these evaporators:

	Helena.	*Wilmington.*
Duration of test, hours	2	4
Pressure in coils, pounds per square inch	52.78	67
shell, pounds per square inch	23.5	23
Temperature distilled water, degrees Fahrenheit	90	86
Concentration	$\frac{2.5}{32}$	$\frac{2.7}{32}$
Water distilled, gallons	261.5	540
per hour, gallons	130.75	135
rate per day, gallons	3,138.	3,240

The water was good. Each evaporator was run for 30 minutes before test was commenced.

Ash Hoists.—There are two ash hoists, one for each fire room, as auxiliaries to the See ash ejector. The ash hoists are single cylinder, direct pull engines, with a traveling block, hoisting by a continuous wire rope.

Forced Draft.—The forced draft is supplied by two 42-inch Sturtevant blowers in each fireroom, drawing air from the ventilators and delivering it into the closed fireroom. The fans are driven by enclosed double engines with cranks at 180 degrees. The diameter of the cylinders is $3\frac{1}{2}$ inches and the stroke, $2\frac{1}{2}$ inches.

Electric Plant.—The lighting installation of each ship consists of 239 incandescent lamps, one search light and one set of signaling apparatus. There are seven electric mains supplied by seven feeders. All the wires running throughout the engine and fire rooms are protected by iron tubing made water and steam tight.

The *Wilmington* is supplied with one dynamo of 200 and one of 100 amperes, each at 80 volts. They were built by the General Electric Company and are of the compound-wound, quadripolar, direct-connected type. The *Helena* has two 100-ampere dynamos of the same type and make.

All dynamos are driven by double vertical engines, the cylinders of the 8-kilowatt dynamos being $6\frac{1}{2}$, and those of the 16-kilowatt dynamo, $8\frac{1}{2}$ inches in diameter. The stroke of both sizes is 5 inches.

Ventilation.—On the *Helena* there is one steam Sturtevant blower, arranged to work either on the exhaust or supply system. This blower is in the port engineer's storeroom.

On the *Wilmington* there are nine small ventilating fans, placed in the different compartments, and driven by electric motors. The air ducts are thus short and confined to each compartment, instead of the long main duct with branches required by the single large fan, as in the *Helena.*

Ice Machine.—There is fitted an Allen dense air ice machine of one ton capacity. There are also the usual fittings to cold storage rooms and scuttle butt. The following test was made on the ice machine for the *Helena:*

Duration of test, hours	5
Steam pressure, pounds per square inch	92.9
Revolutions, per minute	146.9
Pressure in air compressing cylinder	228.9
expanding cylinder	60.6
Temperature, refrigerating compartments *	5
drinking water	38.6
water in tanks	52
atmosphere	68.4
Pounds of ice made	172

BOILERS.

There are six single-ended, horizontal, cylindrical, return fire tube boilers, with one furnace each. They are built of two courses of two sheets each, and the tops of the front and back heads are curved.

The usual mountings are fitted, except the hydrokineter, instead of which there is a feed water circulator acting on the same principle as a suction tee. This fitting is placed in connection with the internal feed pipe and mixes the water from the bottom of the boiler with the entering feed.

DATA FOR ONE BOILER.

Shell, length over heads, feet and inches	9–9⅜
diameter, mean, feet and inches	7–9⅜
thickness, inch	1⅛

* Door was opened several times during test.

Shell, number of courses	2
sheets in course	2
Furnaces, number	1
diameter (least inside), inches	42
(greatest inside), inches	44 13/16
thickness, inch	19/32
length of grates, inches	72
number of grate bars each furnace	28
Tubes, number of ordinary No. 10, B. W. G.	119
stay No. 6, B. W. G.	39
diameter all tubes outside, inches	2¼
length between tube sheets, inches	85½
center to center of tubes vertically, inches	3¼
horizontally, inches	3¼
internal diameter ordinary, inches	1.982
stay, inches	1.844
thickness of tube sheets, inch	5/8
Thickness of heads, top, inch	13/16
bottom, inch	5/8
Tube heating surface, square feet	663
Plate heating surface, furnaces, square feet	50
combustion chambers, square feet	87
Total heating surface, square feet	800
Grate surface, square feet	21
Ratio G.S. to H.S.	1:38.1
Area through tubes, square feet	3.27
Steam space, normal, cubic feet	98.3

TOTALS FOR SIX BOILERS.

Heating surface, square feet	4,800
Grate surface, square feet	126
Area through tubes, square feet	19.64
Volume steam space, cubic feet	589.8
Area water surface, square feet	370.5
Net area one main smoke pipe, square feet	21.64

RATIOS.

Total G.S. to H.S.	1 to 38.1
Area through tubes to G.S.	1 to 6.42

Coal Bunkers.—There are ten coal bunkers, as shown on plan, the total capacity of which, based on 42 cubic feet to the ton, is 300 tons. The capacity and cubical space of each bunker is shown in the following table.

	Starboard.		*Port.*	
	cu. ft.	tons.	cu. ft.	tons.
Forward thwartship bunkers	1,050	25	924	22
fore and aft bunkers	1,344	32	1,260	30
After thwartship bunkers	1,890	45	1,722	41
fore and aft bunkers	1,470	35	1,470	35
Bunkers abreast engine room	714	17	756	18
Totals		154		146

Weights.—The following weights may be used to show the relative weight and power.

Weight of engines proper, inclusive of bed plate, framing, valve gear, valves, crank shafts, cylinders, hand gear, relief valves, receiver pipes, lagging and air pump, tons	20.045
Condenser, with main exhaust pipe, air pump suction and discharge, tons	4.881
One main boiler, empty, without fittings, tons	10.841
Fittings and lagging for one boiler, tons	1.273
Water in one boiler (4 inches above connection), tons	5.416
Main circulating pump, engine, injection pipe and outboard delivery pipe (not including valves), tons	0.944

THE TRIALS.

The official trial of the *Wilmington* was run on Saturday, the 27th of March, and that of the *Helena* on the following Monday, the 29th of March, 1897.

The *Wilmington* left the ship yard on March 20 and the *Helena*, on March 21. Both vessels steamed under natural draft to New London showing excellent results. On the run of the *Helena* from Newport News to New London, she made 110 nautical miles in 8 hours and 19 minutes, being an average of 13.22 knots under natural draft and with no attempt to urge the fires. The distance was measured by both patent log and by bearings from the chart.

For the official trial, a course of 27 nautical miles was laid out from a spot off Horton Point, Long Island, to one opposite Oldfield Point, the average depth being about 18 fathoms.

The *Wilmington* started over the line at 9 hours 4 minutes 38 seconds, A. M., and crossed the last buoy at 10 hours 57 minutes 29.5 seconds. On the return run, she crossed the line at 11 hours 4 minutes 20.5 seconds, A. M., and finished at 12 hours 47 minutes 23 seconds, P. M., making the total elapsed time 3 hours 35 minutes 54 seconds.

The *Helena's* run was as follows: Crossed the line at 8 hours 24 minutes 24.5 seconds, and the end buoy, at 10 hours 10 minutes 1 second A. M.; on the return run, crossed the line at 10 hours 42 minutes 25 seconds, and finished at 12 hours 21 minutes 50.5 seconds. The total elapsed time was 3 hours 25 minutes 2 seconds. The mean corrected speed of the *Wilmington* was 15.0756 knots, and for the *Helena*, 15.49712 knots. This gained for the contractors a bonus of $41,512 for the first, and $49,942 for the second.

Both trials passed off without the slightest mishap, the bearings were cool without water on them, the boilers steamed steadily and the air pressure was maintained easily. The engines could have worked off more steam had the boilers furnished it, but only to a very slight extent. The coal burned was selected New River steam coal, and all boilers were in use.

The trial crew of each boat numbered 91 officers and men, including pilots and draughtsmen. The engine room force, 50 in number, was under Chief Engineer J. A. Williams, connected with the company's works, and consisted of the following: Two chief engineers, 4 assistant engineers, 2 deck engineers, 5 special machinists, 2 water tenders, 11 oilers, 11 firemen, 4 coal passers, 2 boiler makers, 2 electricians and oilers, 4 engine draughtsmen and 1 storekeeper. All the oilers were machinists who had worked on the erection of the machinery in the vessels, and the water tenders were leading machinists who had had charge of putting in the fittings.

There were 18 men in the steward's department.

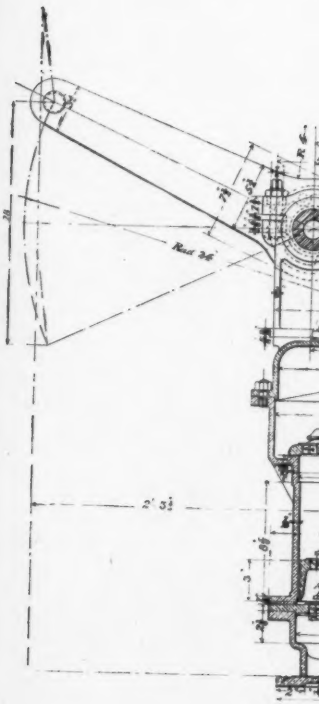
CONTRACT TRIAL DATA.

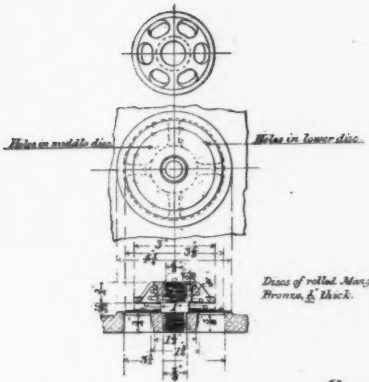
	Helena.	*Wilmington.*
Draught of water, beginning of trial, forward, feet and inches...	8–2½	8–3
aft, feet and inches.....	9–2	9–1
end of trial, forward, feet and inches.....	8–1	8–2
aft, feet and inches.....	9–1	9–1
mean, forward, feet and inches.....	8–1¾	8–2½
aft, feet and inches.....	9–1½	9–1
average on trial, feet and inches.....	8–7⅛	8–7¾
Displacement (mean), tons.....	1,340	1,342
Area immersed midship section.....	327.1	327.5

	Helena.	*Wilmington.*
Wetted surface, square feet	10,257	10,236
Speed, mean, knots	15.497	15.076
Slip, (mean of both screws,) per cent	21.04	21.36
Speed^3 × area immersed midship section ÷ I.H.P. (*A*)	632.3	602.8
Speed^3 × displacement$^{\frac{2}{3}}$ ÷ I.H.P. (*A*)	231.6	223.9
I.H.P. (*A*) per 100 square feet wetted surface	19.04	18.19

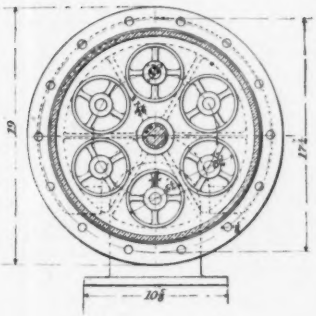
SYNOPSIS OF STEAM LOG.

	Helena.		*Wilmington.*	
	Starbd.	Port.	Starbd.	Port.
Revolutions main engines, per minute	280.02	279.89	272.82	273.31
mean	279.95		273.06	
Piston speed, feet per minute	839 86		819.19	
Steam pressure, in boilers, per gauge	180.9		185.1	
at engines, per gauge	178.7		181.3	
1st receiver, absolute	75.5	70.4	70.2	62.4
2d receiver, absolute	27.2	30.8	30.	24.2
Vacuum in condenser, inches	23.8		24.2	
Throttle valve opening	Wide.		Wide.	
Double strokes of circulating pump per minute	275.2		211.6	
Double strokes of feed pumps, average, per minute	38.7		32.2	
Double strokes of fire and bilge pumps, average, per minute	135.7		73	
Temperature in degrees Fahrenheit:				
Engine rooms	75		74	
Injection	38		37	
Discharge	112		111	
Feed	135		138	
Fire rooms	116		111	
Air pressure in fire rooms, inches of water	1		1	
Revolutions of forced draft blowers, average, per minute	496.9		514.2	
Mean pressure in cylinders, H.P.	80.15	83.65	87.75	84.
I.P.	32.77	30.06	31.38	28.34
L.P.	13.86	15.65	15.70	12.95
Aggregate equivalent pressure referred to L. P. piston	44.82	43.84	45.29	40.50
I.H.P.:				
H.P. cylinder	329.75	344.04	327.47	337.38
I.P. cylinder	314.25	288.07	293.18	265.32
L.P. cylinder	314.44	354.81	347.13	286.83
Collective, each main engine	958.44	986.92	967.78	889.53


Rad 26
2' 3½

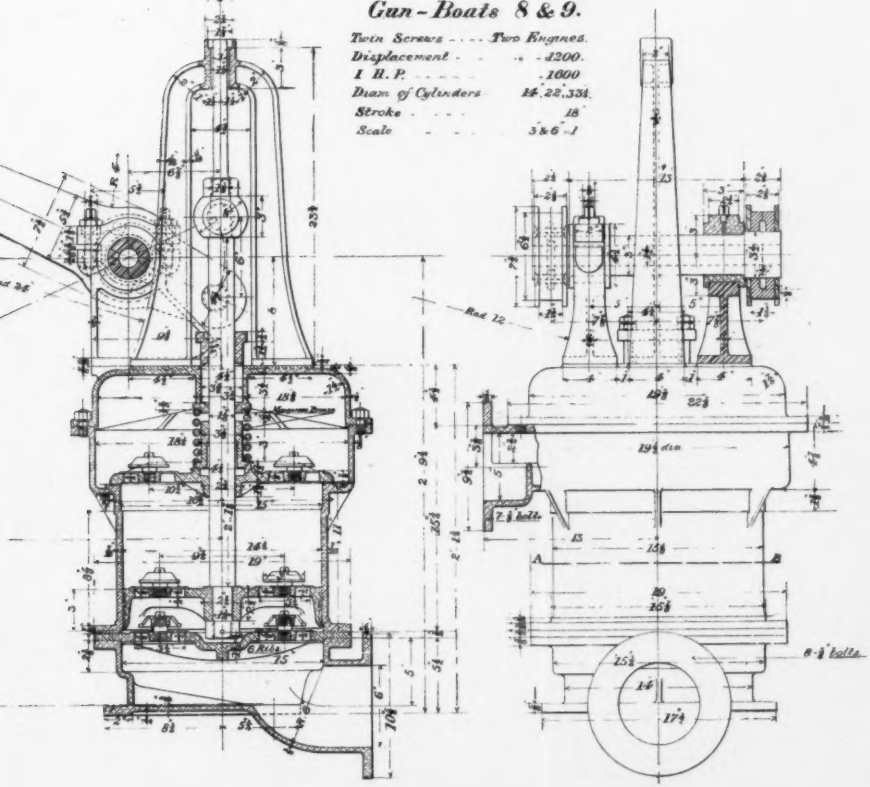


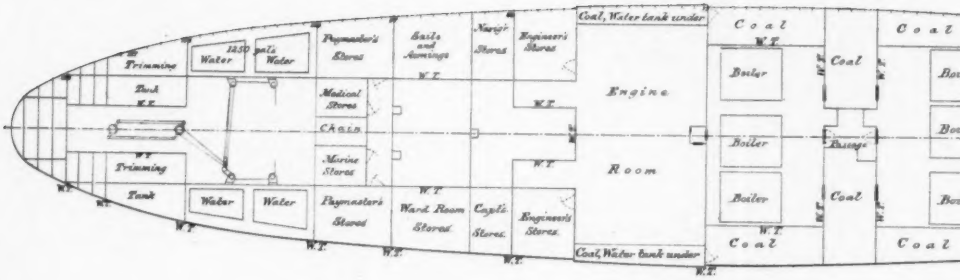
Section A-B.

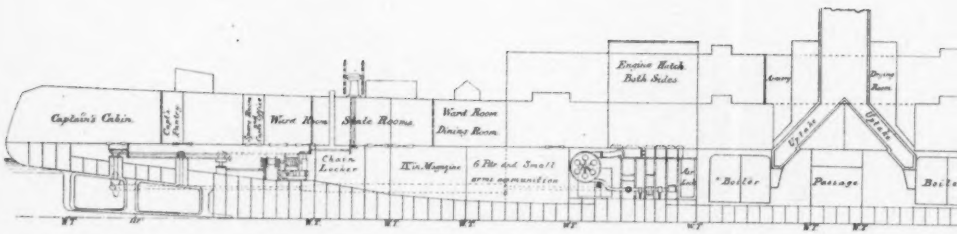


Gun-Boats 8 & 9.

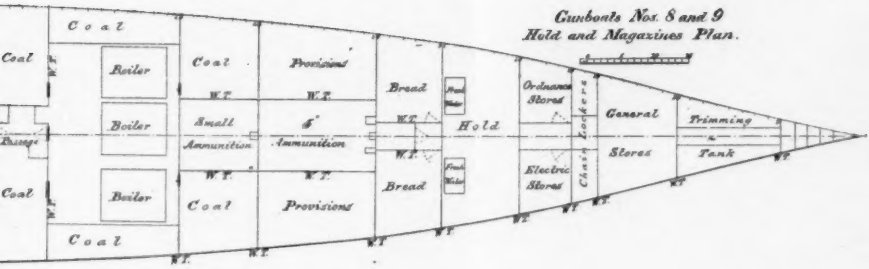
Twin Screws - - - -	Two Engines.
Displacement - -	- 1200.
I. H. P. - - - -	1000
Diam. of Cylinders	14. 22. 33½
Stroke - - - -	18
Scale - - -	3 & 6" - 1

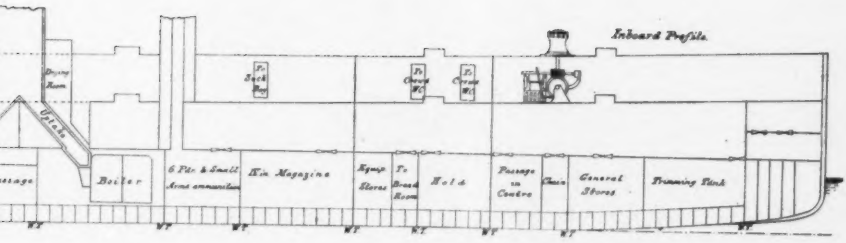



Trimming Tank
1250 gals Water
Water
Paymaster's Stores
Sails and Awnings
Navig' Stores
Engineer's Stores
Coal, Water tank under
Coal
Medical Stores
Chain
Marine Stores
Engine Room
Boiler
Passage
Trimming Tank
Water
Water
Paymaster's Stores
Ward Room Stores
Cap'ls Stores
Engineer's Stores
Coal, Water tank under
Coal
W.T.


Captain's Cabin
Ward Room
Store Rooms
Ward Room Dining Room
Engine Hatch Both Sides
Chain Locker
Magazine
6 Pdr and Small arms ammunition
Boiler
Passage
Boiler
W.T.

Gunboats Nos. 8 and 9
Hold and Magazines Plan.





	Helena.		*Wilmington.*	
I.H.P.:	Starbd.	Port.	Starbd.	Port.
Collective, both main engines	1,945.36		1,857.31	
Circulating pumps	7.8		4.2	
Main feed pumps	5.79		6.65	
Four forced draft blowers	8.4		9.4	
Other auxiliaries	20.83		16.81	
Collective main engines and circulating pumps (*A*)	1,953.16		1,861.51	
Total all machinery in use	1,988.18		1,894.37	
Indicated thrust (main engines only), pounds	15,909.7	16,379.7	16,774.8	15,074.2
Indicated thrust per square foot of developed blade area, pounds	850.3	875.9	897.05	806.1
Indicated thrust per square inch of thrust bearing	50.5	52.0	53.25	47.8
Cubic feet swept per minute by L.P. piston per I.H.P.	102.84	105.89	103.84	95.44
Ratio of volume swept by L.P. piston to that of one air pump bucket per minute	15.7 to 1		15.7 to 1	
Square feet cooling surface per I.H.P. (*A*)	1.15		1.21	
Square feet of H.S. per total I.H.P.	2.41		2.53	
I.H.P. per square foot grate surface, total,	15.78		15.03	
ton machinery (penalty weight)	9.84		9.36	

THE APPLICATION OF THE COMPOUND STEAM TURBINE TO THE PURPOSE OF MARINE PROPULSION.

By the Hon. Charles Parsons.

(Read before the Institution of Naval Architects, April, 1897.)

It has been suggested by Sir W. H. White that a paper giving some account of the application of the compound steam turbine to the purposes of marine propulsion might be of interest to the members of this Institution.

The date of this paper is perhaps somewhat premature, as the *Turbinia*, the first boat fitted with turbine engines, has not as yet completed her experimental trials, but as the results so far ascertained are in some respects remarkable, this perhaps may afford some excuse for their publication.

The manufacture of the compound steam turbine was first commenced in the year 1885 with the construction of small engines for the driving of dynamos; successive improvements were made, and larger engines constructed, but up to the year 1892, the consumption of steam was not such as to justify the application of this class of engine to the purpose of marine propulsion, though on account of its light weight, small size, and high speed of revolution, it presented great advantages over ordinary engines for certain classes of work.

In the year 1892, however, a highly developed compound turbine, adapted for condensing, was constructed for the Cambridge Electric Supply Company, and when tested by Professor Ewing, F.R.S., showed a consumption of steam equivalent to 15.1 pounds per I.H.P. per hour, the boiler pressure being 100 pounds, and the steam superheated to 127 degrees Fahr. above the point of saturation.

More recently, compound turbine engines have been constructed up to 900 H.P., both condensing and non-condensing, and consumptions of steam as low as 14 pounds per I.H.P. with saturated steam, and 100 pounds boiler pressure, have been ascertained in engines of 200 H.P., and still lower consumptions in engines of larger size. Many of the original engines are still doing good work; some, especially the larger sizes of 500 H.P. and upwards, are frequently kept at work for several weeks without stopping. The returns of the Newcastle and District Electric Lighting Company show a yearly cost of up-keep of 2½ per cent. per annum, and the total horse power of turbines now at work in England exceeds 30,000 H.P.

In January, 1894, a syndicate was formed to test thoroughly the application of the compound steam turbine to marine propulsion, and a boat was designed for this purpose. In view of the large amount of alteration that would probably be required before a satisfactory issue was reached, and the large amount of time and expense necessarily involved, it was decided to keep the dimensions as small as possible, but not so small as to preclude the possibility of reaching an unprecedented rate of speed should all the parts work as satisfactorily as was anticipated.

The fulfillment of these anticipations was, however, much delayed, and almost frustrated, by a difficulty which, though foreseen, proved to be of a much more serious character than was anticipated. This difficulty was that termed by Mr. R. E. Froude "The cavitation of the water," or, in other words, the hollowing out of vacuous spaces by the blade of the screw, and this pitfall for the designers of screws for very fast vessels, though indicated by theory to exist, came upon us in the case of our very fast running screw, taxed beyond the usual extent, in its most aggravated form. When the boat and machinery were designed, the trials of the *Daring*, which first drew attention to this difficulty, had not taken place.

The *Turbinia*—as the boat is named—is 100 feet in length, 9 feet beam, and 44½ tons displacement. The original turbine engine fitted in her was designed to develop upwards of 1,500 actual horse power at a speed of 2,500 revolutions per minute.

The boiler is of the water tube type for 225 pounds per square inch working pressure, with large steam space, and large return water legs, and with a total heating surface of 1,100 square feet, and a grate surface of 42 square feet; two firing doors are provided, one at each end. The stokeholds are closed, and the draft furnished by a fan coupled directly to the engine shaft. The condenser is of large size, having 4,200 square feet of cooling surface; the circulating water is fed by scoops, which are hinged and reversible, so that a complete reversal of the flow of water can be obtained should the tubes become choked. The auxiliary machinery consists of main air pump and spare air pump, auxiliary circulating pump, main and spare feed pumps, main and spare oil pumps, also the usual bilge ejectors; the fresh water tank and hotwell contain about 250 gallons.

The hull is built of steel plate, of thickness varying from $\frac{3}{16}$ inch in the bottom to $\frac{1}{16}$ inch in the sides near the stern, and is divided into five spaces by watertight bulkheads.

The approximate weights are:

Main engines	3 tons 13 cwt
Total weight of machinery and boiler, screws and shafting, tanks, &c., tons	22
Weight of hull, complete, tons	15
Coal and water, tons	7½
Total displacement, tons	44½

Trials were made with screws of various patterns, but the results were unsatisfactory, and it was apparent that a great loss of power was taking place in the screw.

To investigate the question thoroughly, a spring torsional dynamometer was constructed, and fitted between the engine and screw shaft, measuring the actual torque transmitted. The measurements conclusively proved that the cause of failure lay entirely in the screws, and, with the object of further investigating the character of this waste of power, a series of experiments was made with model two-bladed screws of 2 inches diameter, revolved in a bath of water heated to within a few degrees of the boiling point, and, in order that the model screw should produce analogous results to the real screw, it was arranged that the temperature of

the water and the head of water above the propeller, as well as the speed of revolution, should be such as to closely resemble the actual conditions and forces at work in the real screw, the object in heating the water being to obtain an increased vapor pressure from the water, so as to permit a representation of the conditions with a more moderate and convenient speed of revolution than would otherwise have been necessary.

The screw was illuminated by light from an arc lamp reflected from a revolving mirror attached to the screw shaft, which fell on it at one point only of the revolution, and by this means the shape, form and growth of the cavities could be clearly seen and traced as if stationary. It appeared that a cavity or blister first formed a little behind the leading edge, and near the tip of the blade; then, as the speed of revolution was increased, it enlarged in all directions until, at a speed corresponding to that in the *Turbinia's* propeller, it had grown so as to cover a sector of the screw disc of 90 degrees. When the speed was still further increased, the screw, as a whole, revolved in a cylindrical cavity, from one end of which the blades scraped off layers of solid water, delivering them on to the other. In this extreme case nearly the whole energy of the screw was expended in maintaining this vacuous space. It also appeared that when the cavity had grown to be a little larger than the width of the blade, the leading edge acted like a wedge, the forward side of the edge giving negative thrust.

From these experiments it would appear that in all screws, of whatever slip ratio, there will be a limiting speed of blade, depending upon the slip ratio and the curvature of the back—in other words, on the slip ratio and thickness of blade; beyond this speed a great loss of power will occur; and that should the speed of ships be still further increased, the adoption of somewhat larger pitch ratios than those at present usual will be found desirable.

It is not proposed here to trace further the losses of power by cavitation, but generally speaking, the effect is felt in the case of the real ship, not in the racing of the screw, but in loss of propulsion effect.

In the model experiments, however, in hot water, the effect was both loss of propulsion effect and also racing, as would naturally be expected from the fact of greater vapor density of the water in the latter case rendering the cavities more stable.

A series of model experiments on cavitation in cold water on the lines described would be extremely interesting, and probably instructive, but would require more elaborate, powerful, and extremely high speed apparatus than was at our disposal. It would also seem that the limitation imposed on slip ratio tends in favor of larger pitch ratio for very fast vessels.

The single compound turbine engine was now removed from the boat and replaced by three separate compound turbines, directly coupled to three screw shafts, working in series on the steam, the turbines being the high pressure, intermediate and low pressure, and designed for a complete expansion of the steam of 100-fold, each turbine exerting approximately one-third of the whole power developed, the three new screw shafts being of reduced scantling.

By this change the power delivered to each screw shaft was reduced to one-third, while the division of the engine into three was favorable to the compactness and efficient working of the turbines. The total weight of engines and the speed of revolution remained the same as before. The effect on the screws was to reduce their scantling, and to bring their conditions of working closer to those of ordinary practice. The thrust of the propellers is balanced by steam pressure in the motors.

The rest of the machinery remains the same, though some changes in arrangement were necessary. The usual lignum vitæ bearings are used for the screw shafts. The engine cylinders lie closely to the bottom of the boat, and are bolted directly to small seatings on the frames of sufficient strength to take the thrust of the propellers. The center of gravity of the machinery is consequently much lower than with ordinary engines.

At all speeds the boat travels with an almost complete absence of vibration, and the steady flow of steam to the motors may have some influence on priming; at any rate, no sign of this has

yet occurred with ordinary Newcastle town water. No distilling apparatus has been fitted.

The boat has been run at nearly full speed in rough water, and no evidence of gyroscopic action has been observable, though such a result would be anticipated from the known small amount of these forces under actual conditions; indeed the *Turbinia* has so far proved herself an excellent sea boat.

The oiling of the main engines is carried on automatically under a pressure of 10 pounds per square inch by a small pump worked off the air pump engine; a small independent duplex oil pump is also fitted as standby. The main engines require practically no attendance beyond the regulation of a small amount of live steam to pack the glands and keep the vacuum good.

The advantages claimed for the compound steam turbine over ordinary engines may be summarized as follows:

1. Increased speed.
2. Increased economy of steam.
3. Increased carrying power of vessel.
4. Increased facilities for navigating shallow waters.
5. Increased stability of vessel.
6. Increased safety to machinery for war purposes.
7. Reduced weight of machinery.
8. Reduced space occupied by machinery.
9. Reduced initial cost.
10. Reduced cost of attendance on machinery.
11. Diminished cost of upkeep of machinery.
12. Largely reduced vibration.
13. Reduced size and weight of screw propellers and shafting.

APPENDIX.

TRIALS OF THE "*TURBINIA*."

In December of last year several runs were made on the measured mile, and the maximum mean speed obtained after due allowance for tide was 29.6 knots, the mean revolutions of the engines being 2,550 per minute. Since then new propellers of increased pitch ratio have been fitted.

Further trials were made on April 1.* The mean of the two consecutive runs gave a speed of 31.01 knots, the mean revolutions of the engines being 2,100 per minute, the fastest run being at the rate of 32.61 knots.

The utmost horse power required to drive the boat at the speed of 31.01 knots is 946, as calculated from experiments on her model, made at Heaton Works, on the method of the late Mr. William Froude.

Assuming the ratio of thrust horse power to indicated horse power to be 60 per cent. (which appears to be the ascertained ratio for torpedo boats and ships of fine lines), the equivalent indicated horse power for 31.01 knots is 1,576.

The feed water supplied to the boiler was measured by a Siemens water meter previously calibrated under the working conditions, and found to be substantially correct.

These measurements were made when running at a speed of 28 knots, and the consumption at 31.01 knots has been calculated from these measurements according to the known law between steam pressure and consumption, and by the observed steam pressures on the engines at the respective speeds. The consumption at 31.01 knots is approximately 25,000 pounds per hour, or 15.86 pounds per indicated horse power. It should be observed that the assumption of the thrust horse power being 60 per cent. of the indicated horse power presupposes that the propellers are of the best form attainable, and should those now fitted be superseded by others of higher efficiency, as is possible, and, indeed, probable, then the figures of consumption per indicated horse power will be correspondingly improved, and the speed of the boat increased.

The consumption of steam at 11.4 knots speed has been measured by meter, and found to be 2,700 pounds per hour, or equivalent to a coal consumption of about 24.6 pounds per knot.

* "Engineering," London, says: "We learn that during an official trial of the *Turbinia*, carried out on April 10, a speed of 32¾ knots was attained, this being, we believe, the highest speed yet attained by any vessel." The official trials were made on April 9-14, by Professor Ewing. On the last day, the boat was started from rest to a speed of 28.5 knots in 20 seconds; and brought to rest from the same speed in 35 seconds. The *Turbinia* has nine propellers.

CONDITIONS OF RUNNING OF *TURBINIA* AT 31.01 KNOTS SPEED.

Mean revolutions of engines	2,100
Steam pressure in boiler, pounds	200
at engines, pounds	130
Vacuum at exhaust of engines, pounds	13½
Speed of boat, knots	31.01
Calculated thrust horse power	946
indicated horse power	1,576
Consumption of steam, reduced to basis of 31.01 knots, pounds	25,000
per indicated horse power per hour, pounds	15.86
Total weight of machinery, including boiler, condensers, engines, auxiliaries, shafting, propellers, tanks, water in boiler, and hot well, in working order, tons	22
Indicated horse power per ton of total machinery	72.1

Owing to adverse weather, these trials have been much delayed, and had finally to be made under unfavorable circumstances. They are, however, believed to be substantially accurate.

ABSTRACT OF DISCUSSION FROM "ENGINEERING," LONDON.

Mr. Yarrow opened the discussion by congratulating the author very heartily on the success he had attained by the application of the steam turbine as a motor for vessels. If any one had predicted some years ago that such remarkable results could be obtained with this engine, they would certainly not have received credence. The paper was not one which invited criticism, but admiration, and he specially drew attention to the very ingenious experiment on cavitation which the author had made.

Mr. Thornycroft said that in the paper there was evidence of something which might prove very important in the future. The small steam consumption for the power obtained and the light weight of machinery showed that improvements had been made in the design of steam turbines. The consumption of steam was nearly as small as the smallest record with which he was acquainted, and to get the power stated from so light an engine it was evident that recourse must be had to very fast running. The details bore this out, for the motor ran at 2,100 revolutions a minute. One difficulty, however, remained: how far would this high speed be available? The difficulty of reducing revolu-

tions was very great, and Mr. Parsons was quite right in the effort he had made to drive direct. He would like to ask him, however, how far the speed could be reduced to an approximation of ordinary rates of revolution without unduly detracting from the efficiency of the motor. The experiments described in connection with cavitation were admirable. His firm had experienced difficulty at 400 revolutions per minute. The displacement of the *Turbinia* was small, it would be noticed, and therefore the propellers had to be of small diameter. It would be of interest if the author would state the diameter of the propellers, and it would also be valuable information if he would say what would be the results in future experiments of superheating. When the impact of steam was used instead of having recourse to a tight piston, the greatest difficulty in using superheated steam was overcome; but the most important point that had to be solved was now how far the number of revolutions could be reduced.

Mr. Swinton said that he had made runs in the boat, and could confirm what was said about the absolute absence of vibration; there was also no gyroscopic action, and no difficulty in rough weather; the boat was a good, serviceable sea boat. Mr. Ramage asked if there was any difficulty in reversing the engine. Mr. Henwood wished to know whether the vessel could be run at slow speeds, and whether the boat could be brought alongside a pier with facility. Another speaker asked what use a boat of these dimensions would be, the displacement being only half that of ordinary practice. Mr. Corrie pointed out that the details of the construction were left to the imagination. He thought the author's claims a little premature, because they had had so much difficulty in working a little boat that the machinery was not likely to succeed on a larger scale.

Mr. Mark H. Robinson, in reference to the last speaker's remarks, pointed out that the vessel was experimental, and it was hardly fair to go into the changes that might be required for larger vessels. It should be remembered that the torpedo boat began with the small river yacht, and that torpedo boat construc-

tion had a most important influence on the whole of marine engineering practice. He would be glad to know what was the amount of superheat in the steam; to which the author replied that there was no superheat in this case. Mr. Leslie Robinson said that it was curious that the speed of this boat was the same as that of the *Forban*, which, however, was a bigger vessel; there was also the same grate area. It would, therefore, be evident that the horse power for the square foot of grate would be much higher in the French boat.

Mr. Parsons, in reply, thanked Mr. Yarrow for the very kind remarks which he had made. Mr. Thornycroft's question as to how far the speed of the turbines could be reduced was one which could not be answered in a word, but in ships of larger beam it would be certainly possible to make some reduction in the speed; in fact, the number of revolutions could be lessened in direct proportion as the diameter of the turbine was increased. Down to 15 knots screws could be driven direct; below that gearing would have to be used, and then the turbine would be applicable to all classes of ships. He looked with distrust on gearing; the small gearing might be made to work, but the application of large geared wheels was a questionable device. The steam pressure could be as high as the boiler would give. It was quite possible to use superheated steam; the difficulty was in getting it. If the superheater were put in the uptake, the temperature of which varied widely, according to the state of the fire, &c., or from combustion taking place there, the amount of superheat would also vary greatly. The chief advantage in using the turbine was that, by compounding, the steam could be expanded down to a lower pressure than ordinarily used; he was able to work down to 1 pound final pressure, and at the same time could work with very high initial pressure. That gave a very wide range of expansion and consequent economy. If he were to work up to 300 pounds pressure he would obtain still better results. The boat was driven in a sternward direction by means of a reverse turbine on one of the shafts; the steam was shut off from the ordinary turbine and turned on to the reversing turbine. With regard

to steaming at low speeds there was no difficulty; the engine was small and less steam was put into it. They could go at 11½ knots with 15 pounds pressure absolute, and could run down to any speed desirable. He had not yet been able to test the most economical speed of steaming, partly because the weather had been so bad, but he hoped soon to make efficiency tests. As to the remarks made about the *Forban*, he was unable to see their relevancy. It was said that she had the same grate area as his own boat, but she was three times the displacement, and would, therefore, require a proportionate increase in power to drive her at the same speed; remembering also that she was a longer boat. He had been asked if the steam had any cutting action on the blades, to which he would reply there was no such action. By compounding, the difference in pressure was small in any one element of the motor, and the velocity was, therefore, low; the only deterioration arose from wear, but the clearances were sufficient to put any difficulty from this cause beyond the reach of probability. The engines would develop up to 3,000 horse power without distress; the diameter of the propellers was 18 inches. He found that cavitation began when the pressure on the blades of the propeller exceeded 11¼ pounds. He used normal screws slightly inclined, and had not gone into the refinement of increasing pitch in any one propeller, or in the propellers relatively to each other. The shafts were slightly inclined, so that the after screws worked partly in the wake of those in front of them, and partly in fresh water.

NOTES.

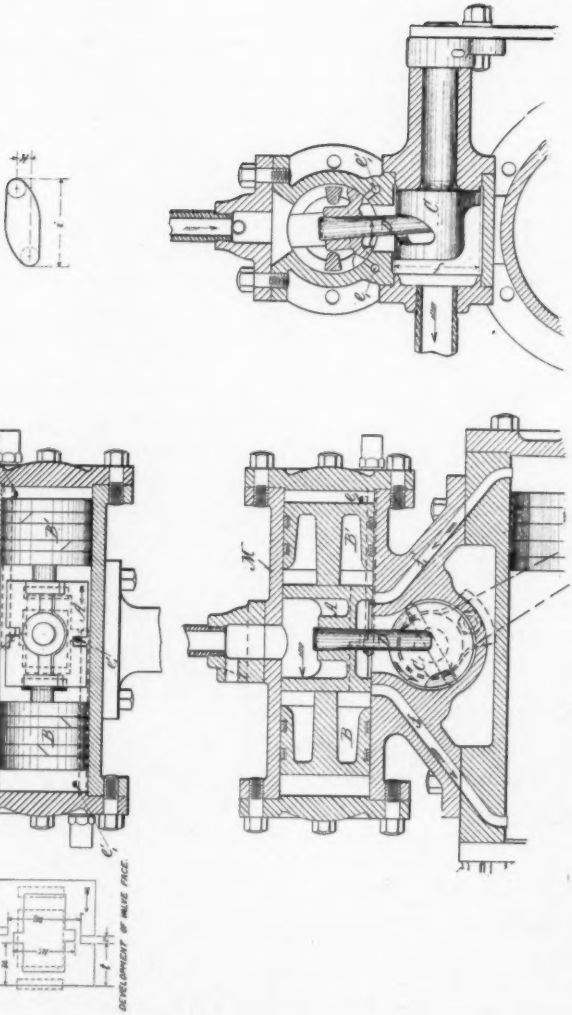
THE DAVIDSON PUMP VALVE GEAR.

The JOURNAL is indebted to Chief Engineer W. N. Little, U. S. N., for the following notes on the above valve gear:

Auxiliary machinery on board ship frequently causes more trouble and requires more attention than the main engines; and not the least troublesome of auxiliaries is the flushing pump. If this latter very important servant be a duplex pump, it will "short stroke" or slam, or both. If it be the single affair with auxiliary steam cylinder valve gearing, the latter will come in for a share of suspicion and adjustment on the occasion of every misbehavior of the pump, when the cause is, perhaps, a weak or broken valve spring or a leaky water valve. Once a month the water valves need examination and probably renewal, for a slight leak of valve by corrosion of seats will cause the pump to slam.

I know of no better all-round pump than the Davidson; and when it fails to work, it is because of long and faithful service, and a need of certain refitting or a renewal of part of its valve mechanism, which can be done on board ship. It is not sufficient, nor always important, to refit accurately the auxiliary piston; yet, the average machinist will begin in that way to remedy the to him unknown defect. If left to his own devices, after consuming several days of valuable time, he may succeed in getting the pump to work, but he cannot tell how long it may be relied upon. Nor in shops ashore are mechanics found more able to diagnose the case of a faulty pump sent to them for treatment, as a recent experience at a navy yard has demonstrated.

As one of the small matters of engineering, usually relegated to machinists, possibly there are members of the association who have given but little if any thought to this subject, and who may find interest in the following analysis of the Davidson pump valve gearing. It is intended to place, in a readily useful form, a means of quickly and accurately deciding upon the repairs or


DEVELOPMENT OF VALVE FACE.

adjustment required. The impressions on paper of the valve, valve seat and cam will readily and accurately supply the measurements of the ports, and, with all the other particulars and data required, or of interest, can be taken by the machinist.

DATA FROM PUMP IN QUESTION.

1 a = Water cylinder, diameter of.
2 b = Steam cylinder, diameter of.
3 c = Stroke of main pistons.
4 d = Length of arm on cam shaft, mean.
5 e = Auxiliary cylinder, diameter of.
6 f = Cam, C, diameter of.
7 g = Distance between axes of auxiliary cylinder and cam shaft.
8 h = Diameter of pin in valve, at contact with cam.
9 i = Circumferential length of cam slot.
10 j = Rotative movement of valve, total, measured at face.
11 k = Rotative throw of cam, total, measured at slot.
12 l = Least effective length of valve pin lever, $= g - f/2$.
13 m = Rotation of valve at face $= k \times e$ divided by $2l$, nearly.
14 p = Lap of main valve.
15 q = Diameter of auxiliary cylinder ports.
16 r = Width of bridge between auxiliary valve ports.
17 s = Distance between extreme edges of main ports divided by 2.
18 s_1 = Distance between inner edges of auxiliary seat ports.

For efficient working:

Item 13, x, should have a value equal to diameter of auxiliary port plus width of bridge between auxiliary ports on valve; *i. e.*, $q + r$.

Item 9, y, should have a value of $2p + h + \frac{ec}{d}$.

Item 11, z, should have a value of $x + m = q + r + m$.

POSITION OF STEAM AND EXHAUST PORTS IN VALVE.

In locating the auxiliary valve ports, it should be remembered that:

(1) The steam and exhaust ports open together.

(2) The exhaust port must be kept open at least until the valve completes its necessary throw.

(3) The auxiliary piston is kept from striking the head by overrunning the port at that end, said port subsequently being uncovered by the endwise movement of the piston by the cam, before the auxiliary port opens for the return stroke.

(4) The auxiliary piston should receive steam to throw the valve in the desired direction at the moment the main cylinder port (at the end toward which valve is moving) is closed, and this auxiliary port opening should be sufficient for leakage of auxiliary piston and to impel the latter.

The proportions of the steam valve and ports are best shown by the cut to the left of the section of the valve and cylinder. The same notation of symbols is observed, and the view is through the valve from the back. The development of the valve face is shown in full lines, and the seat ports in dotted lines.

t	equals	$s - q/2$
u	"	t
v	"	$2/3$ width of main cylinder ports.
w_2	"	s_1
w_2	"	$w_1 + 2q$.

Wear of the cam beyond a certain amount will reduce the rotative throw of valve sufficiently to prevent the auxiliary ports from opening; therefore, this is the place to look for the first cause of non-action of the steam end of pump. The rotation of the valve should, of course, be made equal each side of mid position.

SALT WATER FEED FOR YARROW "EXPRESS" BOILERS.

In order to test the effect of feeding the small tube, or "express" water tube boiler with salt water, Messrs. Yarrow & Co. have made some interesting experiments. These trials were made with a torpedo boat fitted with a Yarrow boiler having 1-inch tubes. The manner of making the trials, and the results are shown in the log given below, which is taken from "Engineering," London.

The "express" type of water tube boiler is much lighter and more compact than either the return tube boiler or the water tube boiler with large tubes; steam can be raised quicker, and the boiler subjected to heavier forcing without danger of damage. It has always been held, however, that the smaller tubes of the express boiler would be quickly stopped up in case salt water should gain access to them, through a leaky condenser or other contingencies which must be provided against. Messrs. Yarrow & Co.'s experiments, made on five consecutive days by running from eight to ten hours each day, tend to disprove this theory. It is not, of course, intended that these boilers, any more than other modern marine boilers, should be used with salt water, but the experiments were made to show that, in case of emergency, the straight tube express boiler may be run for considerable periods of time successfully.

The following is the record from the boat's log. The densities given are in thirty-seconds, and refer to a temperature of about 200 degrees Fahrenheit.

On January 21, 1897, at 8·0 A. M., left yard and proceeded down river. Gravesend was reached at 9·40, when main feed was shut off and feed from sea commenced. Air pump was allowed to discharge overboard. Density of $1\frac{1}{2}$ was obtained by 11·45. Speed of engines at this time, 330 revolutions; steam, 150 pounds. Feed from sea was then shut off, and main feed started. At 12·20 P. M. the speed was increased to 379 revolutions, this being maintained for a period of a quarter of an hour; steam 150, to 155 pounds. Little or no priming took place. At 1·20, supplementary feed from sea to condenser was opened, and scum valve on boiler also opened. Revolutions, 291; steam, 160 pounds.

By 2·0 P. M. density was reduced to $\frac{7}{8}$. Supplementary feed was then stopped, main feed shut off, scum closed, and auxiliary donkey from sea started. When density of $1\frac{1}{2}$ had again been attained, feed from sea was stopped and main pumps started. On return journey, Gravesend was passed at 3·0, the yard being reached at 4·30 P. M.

The boiler had thus been carrying sea water for a period of about seven hours. Maximum density, $1\frac{1}{2}$. Everything worked satisfactorily. Practically no priming. The boiler was opened

out on the morning of January 22. It showed no sign of scale anywhere.

On January 25, at 7·30 A. M., left yard and proceeded down river. Gravesend was reached at 8·55, when main feed was shut off, and feed from sea commenced. Density of 1¾ was obtained by 10·34. Revolutions, 376; steam, 160 pounds. Feed from sea was then shut off and main feed started. At 10·42, speed of engines was increased to 453 revolutions, this being maintained for a period of quarter of an hour; steam, 150 pounds. No priming. At 11·34, supplementary feed was started, and scum valve on boiler opened. Revolutions, 362; steam, 150 pounds.

By 1·0 P. M. density was reduced to ½. Stopped supplementary feed, shut off main feed, closed scum, and started donkey from sea. Revolutions, 350; steam, 155 pounds; density, 1¾. At 2·20, density of 1¾ was again attained. Scum was then opened, donkey feed from sea being continued. Revolutions, 357; steam, 150 pounds. On return journey, Gravesend was passed at 3·42. Feed from sea was then stopped, scum closed, and main feed started. Revolutions, 355; steam, 160 pounds. A density of 2 was at times attained. Yard was reached at 5·30 P. M.

The boiler had thus been carrying sea water for a period of 8½ hours. Maximum density, 2. Everything worked satisfactorily. No priming. The boiler was opened out on the morning of January 26. A very thin coating of white dust showed over lower part of drum and inside of tubes. No scale in bottom pockets.

On January 27, at 7·20 A. M., left yard and proceeded down river. Gravesend was not reached until 10·30. Detained at Rosherville by fog. At 10·30, main feed was shut off, and feed from sea commenced. Density of 2½ was obtained by 12·54 P. M. Scum valve was then opened, feed from sea being continued. Revolutions, 368; steam, 150 pounds.

This treatment was carried on until 3·55. There was some little trouble from priming whenever engine was run at a greater speed than 400 revolutions. For the most part a speed of about 350 revolutions was maintained. Steam, about 150 pounds. Density at times rose above 2⅜. Passing Gravesend at 3·55, on

return journey, feed from sea was stopped, scum closed, and main feed started. Revolutions, 351; steam, 155 pounds. Yard was reached at 5·20 P. M.

The boiler had thus been carrying sea water for a period of about seven hours. Maximum density, $2\frac{6}{8}$. Some trouble from priming, otherwise everything satisfactory. The boiler was opened out on the morning of January 28. It showed a light, fairly uniform scale all over drum, if anything thickest at bottom. The scale in tubes very light. No scale in bottom pockets. The boiler had been cleaned out previously to this trial.

On January 30, at 7·30 A. M., left yard and proceeded down river. Gravesend was reached at 9·25, when main feed was shut off, and feed from sea commenced. Density of $3\frac{1}{2}$ was obtained by 11·55. Scum valve was then opened, feed from sea being continued. Revolutions, 360; steam, 155 pounds. At 12·38 P. M., speed of engines was increased to 430 revolutions; this being maintained for a period of quarter of an hour. Steam, 150 pounds. Some little trouble from priming. At 1·15, shut off feed from sea, closed scum, and started main feed. Revolutions, 348; steam, 145 pounds. At 1·45, speed of engines was increased to 458 revolutions, this being maintained for a period of quarter of an hour. Steam, 150 pounds. No priming.

After this, stopped main feed, started donkey from sea, and opened scum. This treatment was continued until 2·50, when Gravesend was passed on return journey. Feed from sea was then stopped, scum closed, and main feed started. Revolutions, 376; steam, 150 pounds. Yard was reached at 4·40 P. M.

The boiler had thus been carrying sea water for a period of about $7\frac{1}{2}$ hours. Maximum density, $3\frac{3}{4}$. Everything worked satisfactorily. Very little priming. The boiler was opened out on the morning of February 1. There was a very light scale in bottom of drum at after end; none at all at forward end. Scale in tubes light. No scale in bottom pockets. The boiler had been cleaned out previously to this run.

On February 4, at 8·30 A. M., left yard and proceeded down river. Gravesend was reached at 10·0, when main feed was shut off and feed from sea opened. Density of $4\frac{1}{2}$ was obtained by

1·15 P. M. Revolutions, 368; steam, 150 pounds. Scum valve was then opened, feed from sea being continued. This treatment was carried on until 4·10.

For a period of three hours an average speed of over 380 revolutions was maintained, with density of $4\frac{1}{2}$, there being no trouble whatever from priming. Steam varied from 150 pounds to 155 pounds. A density of $4\frac{5}{8}$ was at times attained. At 4·10, Gravesend was passed on return journey. Feed from sea was stopped, scum closed, and main feed started. Yard was reached at 6·0 P. M.

The boiler had thus been carrying sea water for a period of about 8 hours. Maximum density, $4\frac{5}{8}$. Everything worked satisfactorily, there being no priming. The boiler was opened out on the morning of February 5. A light scale, not more than $\frac{1}{32}$ inch thick anywhere, showed all over lower half of drum and inside of tubes. No scale in lower pockets, a slight deposit of mud being there. The scale in drum and tubes seemed to be composed, for the most part, of sand. This was to be expected, as the water between Gravesend and Southend had a very muddy appearance. The scale could very easily be removed. The boiler had been cleaned out previously to this run.

On March 12, at 10·20 A. M., left yard. Gravesend was reached at 12·0 midday. Ten minutes later main feed was stopped, and feed from sea started. Ran down round the Nore and back, passing Gravesend at 2·45 on return journey. A density of almost 3 was reached, and for a period of $2\frac{3}{4}$ hours the engines were run at about 400 revolutions. Steam varied from 150 to 175 pounds. At 2·45, off Gravesend, feed from sea was stopped and main feed started. Boat then proceeded up river, the yard being reached at 4·15 P. M.

The boiler had thus been carrying sea water for a period of about $4\frac{1}{4}$ hours. Everything worked satisfactorily. Little or no priming. The boiler had been cleaned out previously to this trial.

EFFECT OF LONG PIPE CONNECTIONS ON INDICATOR DIAGRAMS.

In a paper read before the American Society of Mechanical Engineers in 1896, Mr. W. F. M. Goss gives the following con-

clusions, derived from numerous experiments made with various lengths of pipe between the cylinder and the indicator:

1. If an indicator is to be relied upon to give a true record of the varying pressures and volumes within an engine cylinder, its connection therewith must be direct and very short.

2. Any pipe connection between an indicator and an engine cylinder is likely to affect the action of the indicator; under ordinary conditions of speed and pressure, a very short length of pipe may produce a measurable effect in the diagram, and a length of three feet or more may be sufficient to render the cards valueless except for rough or approximate work.

3. In general, the effect of the pipe is to retard the pencil action of the indicator attached to it.

4. Other conditions being equal, the effects produced by a pipe between an indicator and an engine cylinder become more pronounced as the speed of the engine is increased.

5. Modifications in the form of the diagram resulting from the presence of a pipe, are proportionally greater for short cut-off cards than for those of longer cut-off, other things being equal.

6. Events of the stroke (cut-off, release, beginning of compression) are recorded, by an indicator attached to a pipe, later than the actual occurrence of the events in the cylinder.

7. As recorded by an indicator attached to a pipe, pressures during the greater part of expansion are higher, and during compression are lower, than the actual pressures existing in the cylinder.

8. The area of diagrams made by an indicator attached to a pipe, may be greater or less than the area of the true card, depending upon the length of the pipe; for lengths such as are ordinarily used, the area of the pipe cards will be greater than that of the true cards.

9. Within limits, the indicated power of the engine is increased by increasing the length of the indicator pipe.

10. Conclusions concerning the character of the expansion or compression curves, or concerning changes in the quality of the mixture in the cylinder during expansion or compression, are unreliable when based upon cards obtained from indicators attached

to the cylinder through the medium of a pipe, even though the pipe is short.

COLLAPSE OF A PAIR OF CORRUGATED FURNACES.

[From "The Locomotive," Hartford, Conn., February, 1897.]

The illustrations represent an accident which recently occurred on the Pacific Coast, and which has several instructive features. It shows, for example, the wonderful amount of deformation that a well made corrugated furnace can undergo, without leading to a disastrous explosion; and it also conveys the useful lesson that unnecessary complications in valves and piping should be carefully avoided, since they are almost sure to result in trouble, at some time or other.

The accident in question took place in one of the four compound marine boilers of the whaleback steamer *City of Everett*, which is used as a freight steamer in the coasting trade between San Diego, Cal., and British Columbia. The *City of Everett* was built of steel in 1894, at Everett, Wash., and is 346 feet in length, 42 feet 8 inches in breadth, and 13 feet 7 inches in depth. She has four compound marine boilers, each 132 inches in diameter and 11 feet long. The shells are of steel, 0.938 of an inch thick, and of 60,000 pounds tensile strength; and the pressure allowed by the Government inspector is 168 pounds per square inch.

Each of the four boilers has two corrugated steel furnaces, 40 inches in diameter and 8 feet 6 inches long. The furnace shown in Fig. 1, is from the forward starboard boiler, both furnaces of which were burned and bulged down by blowing the water out of the boiler while there was a fire in each of them.

The blow-off pipes from the four boilers are connected into cross, or thwartship blow pipes, which pass over the tops of the boilers as shown in Fig. 2, and are provided with valves on the port and starboard sides of the vessel. It is this arrangement of the blow pipes which led to the accident under discussion.

It appears that the vessel was loaded in British Columbia with coal, and after leaving Victoria, B. C., and while steaming across the straits of San Juan de Fuca, the engineer in charge ordered

the water tender, or fireman, to blow down three or four inches of water from the forward starboard boiler. In carrying out this order the fireman opened the valve on the front head of the starboard boiler, and also the outboard blow valve on the starboard side. (This will be understood by reference to Fig. 2.) After

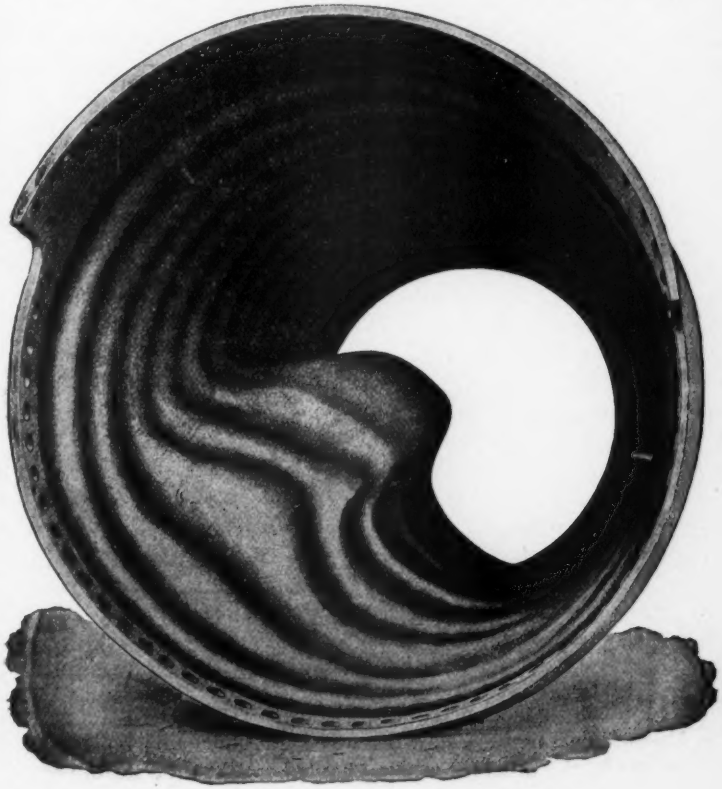


FIG. 1.

blowing down as far as desired, the top outboard valve was closed. This, of course, stopped the blowing at once; but when the man came down from the ladder *he forgot to shut the lower valve, on the head of the boiler.*

When the steamer reached Port Townsend, one of the breeching bolts on the forward port boiler was found to be leaking, and the chief engineer therefore ordered the pressure to be lowered on that boiler, and that water blown out so that the bolt could be renewed. He also gave orders to wash out the two after boilers, and carry steam only on the forward starboard boiler (*i. e.*, the one on the right, in Fig. 2). As the *Everett* was to lie at

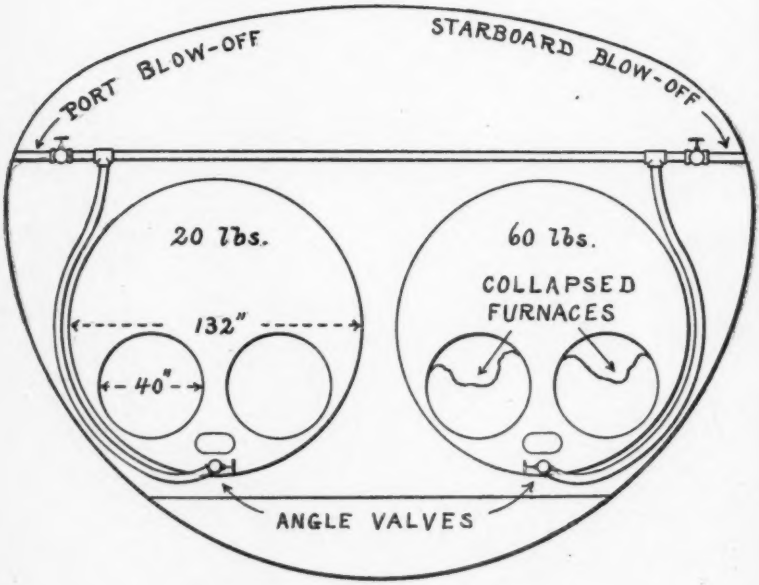


FIG. 2.

the dock over night, a slow fire was kept under this boiler, the fires under the other three being hauled. When the steam pressure on the three cooling boilers had been reduced to about 20 pounds, orders were given to open the blow-off valves attached to them, and blow down. This was done, the blow-off valve on the front head of the starboard boiler being open all this time, although it was supposed to be shut.

The result hardly needs to be told. The boilers from which the fires had been drawn were under only 20 pounds of steam, while the forward starboard boiler, with a fire in each furnace, was carrying 60 pounds, so that it emptied itself much more rapidly than any of the others. After some little time one of the firemen opened one of the doors of the forward starboard boiler, and found the furnace red hot and bulged down, as shown in the engravings. Upon investigation he found the other furnace in this boiler in the same condition. The fires were at once hauled out, and upon examination it was found that the furnaces were down 21 inches, the corrugations being pulled out so that at the bottom of the bulges the furnace was almost smooth; but no signs of fracture could be discovered.

If these furnaces had been poorly made, or if they had been constructed of a material deficient in ductility, it is almost certain that a disastrous explosion would have followed the rough usuage to which they were subjected; and this fact ought to satisfy anyone of the paramount importance in boiler construction of having good material and good workmanship. The unwisdom of giving a fireman too many things to remember will also be apparent; for if each boiler had been provided with its own separate blow pipe, discharging directly into the sea without any connection with the other boilers, this accident could not have happened.

THE BRITISH NAVY ESTIMATES.

[From "Engineering," London.]

On March 2, the Navy Estimates for 1897–98 were presented to Parliament. The total estimates for the financial year, which begins on April 1, 1897, are £21,838,000, without appropriations in aid, which amount to £942,473. This is practically the same sum as was voted last year, the net increase for the coming year being only £15,000.

The chief increase is in Vote 1, which provides for the pay of the *personnel* of the Navy. This year we have the total numbers brought to 50 over the round 100,000; that is to say, 6300 more

than last year. The total amount required for payment of these "officers, seamen, boys, coastguard and Royal Marines" will be £4,696,000; which is a little more than a quarter of a million in excess of the last estimates.

Vote 8, which provides for shipbuilding and engineering of warships, is a decrease on last year's expenditure by about half a million of pounds. It should be stated in explanation of the decrease in the shipbuilding vote, that when the programme of building for the three years commencing 1896–97 was settled last February, the sum of £600,000 was transferred from 1897–98 to 1896–97 in order to enable more rapid progress to be made.

Mr. Goschen says in the annual statement that "a grade of engineer officers of warrant rank has been established as an encouragement to engine room artificers who have rendered meritorious service, and at the same time as a step which will contribute towards meeting the growing requirements of the Fleet;" and that "it has been decided to gradually raise the age of entering naval cadets by one year, and to shorten their course of instruction in the Britannia to about 16 months." A few changes are to be made in the Royal Naval Reserve: The executive officers' list is to be increased by over 100, bringing the total to 1,400, besides which there are to be 300 engineer officers. In order that the latter may acquire a knowledge of the working of engines and boilers used in Her Majesty's ships, arrangements are being made to receive a limited number of them for a course of instruction at the home ports. "Opportunities will be given to them to study the construction and repair of machinery in the dockyards, and also the working of the engines and boilers of torpedo boat destroyers and torpedo boats, and of other classes of ships, when undergoing their trials."

We now come to the First Lord's statement referring to "New Construction;" in spite of the varied nature of the work in progress, and the novelty of a large part of it, "on the whole, expectations have been realized in both the dockyard and the contract sections of the work." One of the chief difficulties experienced was that of getting tubes for the water tube boilers required for the destroyers; and it is strange how the bicycle

mania did much to retard the work on Her Majesty's vessels which were fitted with small tube boilers. Retarded delivery of boiler tubes has kept the *Diadem* class back somewhat, but it is estimated that the vessels will be ready within the specified time notwithstanding.

Mr. Goschen says: "Progress on the 30-knot destroyers has not been so rapid as was originally anticipated by the firms carrying out their construction, all of whom had successfully fulfilled the conditions of their earlier contracts for destroyers. Experience has, in fact, proved that with each successive increase in speed, new and unforeseen difficulties have to be surmounted, but there is every reason for anticipating that success will be finally achieved in all the vessels. During the coming financial year all the vessels of this type should be delivered and tried."

The programme of new construction for the coming year includes four battleships, three third class cruisers, two sloops, four twin screw gunboats and two torpedo boat destroyers. A new yacht is also to be built for the Queen at Pembroke. Three of the battleships, the three cruisers and two sloops will be built at the dockyards; the other vessels will be given out to contract. The propelling machinery for all except the two third class cruisers and one of the sloops will be ordered from private firms. No particulars are given of these vessels.

The report concludes this section by a summary of the vessels under construction or completing during the course of 1897–8: Fourteen battleships, eight first class cruisers, nine second class cruisers, ten third class cruisers, two sloops, four twin screw gunboats, fifty-two torpedo boat destroyers, eight light draught steamers for special service and one Royal yacht. The total number of vessels of all classes under construction during the year will be 108. Their aggregate displacement tonnage will be about 380,000 tons, and the aggregate indicated horse power about 800,000. It is anticipated that during the next financial year sixty-six of these vessels will be completed for service, including fifty torpedo boat destroyers.

SHIPS.

UNITED STATES.

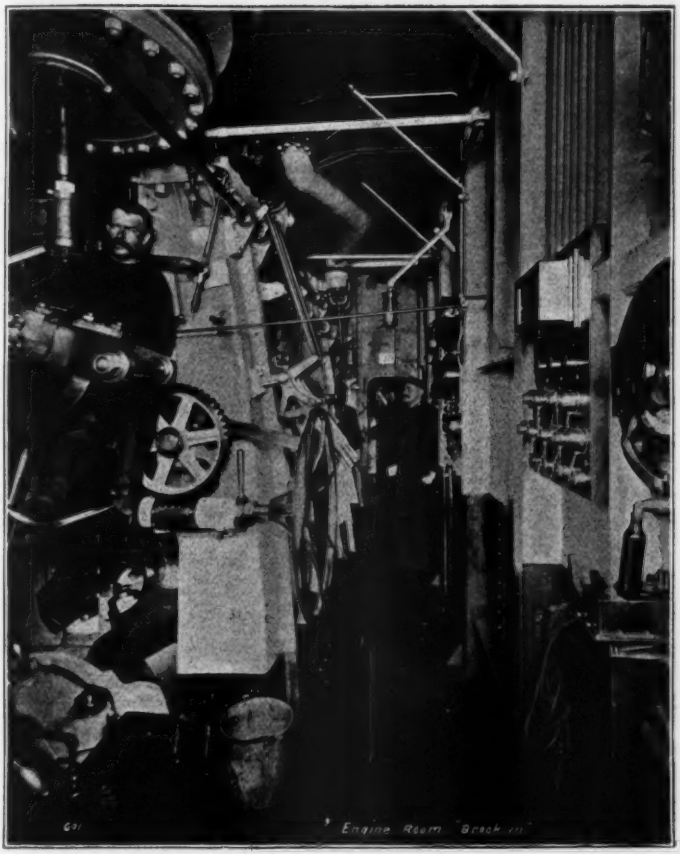
Iowa.—This sea-going battleship (see page 383 of Volume VIII), successfully completed her official contract trial on April 7, running twice over the 33-mile course off the New England coast. As the vessel and the trial will be described in full in the August JOURNAL, the principal trial results only are given now. Mean displacement on trial, 11,358 tons; steam pressure in boilers, 161 pounds per square inch; vacuum, 24.1 inches; revolutions per minute, starboard, 108.61, port, 110.51; I.H.P., starboard, 5,949.18, port, 5,885.3; total, 11,834.48; I.H.P., main engines, air, circulating and feed pumps, 11,933.13.

The first run over the course was northward and was made in 1 hour 57 minutes and 21 seconds, and the second run, southward, in 1 hour 55 minutes and 36 seconds. With tidal corrections, the distance steamed through the water was 66.3407 nautical miles, which gives a true mean speed of 17.0871 knots. Weather and other conditions were favorable and the machinery worked well.

Brooklyn.—This armored cruiser, fully described on page 741 of Volume VIII, successfully made her final acceptance trial of 48 hours on April 27 to 29, a short time after finishing the repairs necessitated by her unfortunate grounding. The results of the full power, natural draft trial, which was part of the whole run and made on April 28, are as follows:

Mean draught of vessel, 23 feet 11 inches; corresponding displacement, 9,180 tons; steam pressure, 148 pounds per square inch; revolutions per minute, starboard, 110.7, port, 110.1; I.H.P., 8,293.7; speed by patent log, checked by revolutions of engines, 18 knots.

The two half tones of one of the engine and fire rooms of this cruiser were reproduced from excellent flash light photographs



FROM A PHOTOGRAPH BY HART, BROOKLYN, N. Y.

FROM A PHOTOGRAPH BY HART, BROOKLYN, N. Y.

taken by Mr. Hart of Brooklyn, N. Y., and kindly sent to the JOURNAL for reproduction.

Marietta and **Wheeling.**—These twin screw gunboats, referred to on page 799 of Vol. VIII, were launched on March 18 by the Union Iron Works, San Francisco, Cal.

Annapolis.—This composite, single screw gunboat, built at the Crescent Shipyard, Elizabethport, N. J., (see page 140 of the February number), successfully completed the four hours' official trial over the course in Long Island Sound on April 22. As the trial and the vessel will be described in full in the August JOURNAL, only the principal results of the trial are here given.

Steam pressure in boilers, 225, at engines, 217 pounds per square inch; vacuum, 24 inches; revolutions per minute, 146.76; I.H.P. of main engine, air, circulating and feed pumps, 1,222.56; speed, corrected, 13.172 knots.

Foote, Rodgers and **Winslow.**—These torpedo boats, building by the Columbian Iron Works and Dry Dock Co., Baltimore, and described on page 799 of Vol. VIII, were formerly known as *Torpedo Boats Nos. 3, 4* and *5*, respectively.

Dupont, Porter and **Rowan.**—These torpedo boats, some particulars of which were given on page 805 of Volume VII, were formerly known as *Torpedo Boats Nos. 7, 6* and *8*, respectively.

The *Porter* made her official trial on February 10, over a measured course of 11.972 nautical miles in Narragansett Bay. Five runs were made over this course, the mean speed, corrected for tidal observations, being 28.63 knots. The mean draught at beginning of trial was 4 feet 9.5 inches. As the JOURNAL hopes to fully describe the *Porter* and *Dupont* in the next issue, only the principal results of the *Porter's* trial are given now.

Steam pressure in boilers, 196 pounds per square inch; vacuum, 23.25 inches; revolutions per minute, starboard, 389.95, port, 389.2; air pressure, 2.2 inches.

The *Dupont* was successfully launched on March 30, by the Herreshoff Manufacturing Co., Bristol, R. I.

Dahlgren, Farragut and **T. A. Craven.**—These 30-knot torpedo boats, described on page 800 of Volume VIII, were formerly known as *Torpedo Boats 9, 11* and *10*, respectively.

Davis, Fox, Gwin, Mackenzie, McKee, Morris and **Talbot.**—These torpedo boats, described on page 801 of Volume VIII, were formerly known as *Torpedo Boats Nos. 12, 13, 16, 17, 18, 14* and *15*, respectively.

30-Knot Torpedo Boats Nos. 19, 20 and **21.**—On April 14, the Navy Department issued a circular defining the chief characteristics of the boats allowed by the Act of Congress, approved March 3, 1897. This act provided for the construction by contract of "not more than three torpedo boats, to have a speed of not less than thirty knots, to cost in all not exceeding $800,000. And not more than two of said torpedo boats shall be built in one yard or by one contracting party. . . . " They are to be built under the provisions of the Act of August 3, 1886, except that no premiums are offered, and all parts must be of domestic manufacture.

Two boats must not be less than 230 tons, and one, not less than 260 tons displacement, the former carrying a trial weight of 41.6 tons each, and the latter, 48.5 tons, in addition to completed hull, machinery and spare parts, with the water necessary for trial. In case bidders are unwilling to carry, on trial, the above trial weights, they must specify in their proposals the additional weights that they propose to carry on trial. In awarding the contracts, however, other things being equal, preference will be given to designs proposing to carry the full weights desired.

The average speed to be maintained for two hours, under conditions to be prescribed by the Secretary of the Navy, must not be less than 30 knots. The trial to be made over a course not less than 30 nautical miles in length.

The boats are to be built and fitted complete for sea by the contractors, with the exception of sea stores, ordnance and ordnance outfit, which, however, will be fitted and secured in place by the contractors. Two conning towers will be placed in convenient positions, one forward and the other aft. Accommodations will be provided for three commissioned officers and an adequate crew. An incandescent electric lighting plant must be

fitted not to exceed 3,300 pounds in weight. No search light will be required.

The battery for the vessels of not less than 230 tons displacement will consist of not less than four 6-pounder R.F. guns and two 18-inch Whitehead torpedo tubes with four torpedoes. The 260-ton vessel will carry at least six 6-pounder R.F. guns and the same torpedo outfit.

The main engines may be either of the triple or the quadruple expansion type. The condenser tubes will be $\frac{5}{8}$ inch outside diameter, No. 18 B. W. G. thick, made of Muntz metal compounded of new metals.

The boilers will be of the water tube type, fitted with approved feed-regulating apparatus, automatic self-closing ash pit doors, and all the necessary valves, pipes and fittings, and be provided with ample facilities for cleaning and repair. The boiler tubes to be of charcoal iron or steel; tubes 1 inch in diameter will be No. 12, and those $1\frac{1}{8}$ or $1\frac{1}{4}$ inches in diameter, No. 11 B. W. G. If of iron, they must be made of the best quality of knobbled, hammered charcoal iron.

The screw propellers will be of manganese bronze, or equivalent metal.

Fresh water tanks of at least 1,500 gallons capacity will be provided for holding fresh water for the boilers. The distilling apparatus will consist of an evaporator having a capacity of 3,000 gallons of water per 24 hours when supplying water for the boilers. A distiller of approved type and a capacity of 800 gallons in 24 hours, at a temperature of not more than 90° F. when the condensing water is taken at 60° F., will be fitted for making drinking water.

All copper and brass piping of and less than 6 inches diameter will be seamless-drawn. All copper pipes not seamless-drawn will be brazed.

The outside appearance of all of the smoke pipes must be the same.

Golden Gate.—This steel revenue cutter, built by Messrs. Moran Bros. Co., Seattle, Wash., was successfully launched on February 13, and made her official trial in Puget Sound in April.

The mean speed for the six hours' run was 13.13 knots. Some particulars of this vessel were given on page 566 of Volume VII.

Revenue Cutters Nos. 4 and 5.—The Globe Iron Works Co. of Cleveland, Ohio, closed a contract on March 31 for constructing these two revenue cutters. These vessels are to be duplicates in all respects, and each will be of about 850 tons displacement. The principal dimensions are as follows: Length over all, 205.5 feet, between perpendiculars, 188 feet; breadth of beam, molded, 32 feet; depth of hold, 17 feet; draught, mean, 10 feet 2¼ inches.

The contract price for each vessel is $193,800, and the total appropriation for each, $200,000. They will be constructed of mild open hearth steel, which must have a tensile strength of not less than 55,000 pounds per square inch, and an elongation of not less than 25 per cent. in a length of 8 inches. Below the water line the hulls are practically duplicates of the *Gresham*, described in this number of the JOURNAL. While the latter vessel is flush-decked and fitted with deck houses, the new cutters will have poop and forecastle decks, and the main deck will be flush. They will be schooner-rigged with pole masts, and carry a small spread of sail. The quarters for the commanding officer will be on the main deck, and those for the ward room officers on the berth deck, occupying all the space between the engine room and the stern of the vessel. Each vessel will be fitted with a torpedo port, and will carry one 6-pounder rapid fire gun, mounted on the forecastle deck.

The propelling machinery will consist of one vertical direct-acting, triple expansion engine, having cylinders 25, 37.5 and 56.25 inches in diameter, and a common stroke of 30 inches. The air pumps will be independent, and of the twin vertical type. The circulating pump is also independent, and of the centrifugal type. Steam will be supplied by four single-ended Scotch boilers, each 11 feet 8 inches outside diameter and 10 feet long. Each boiler will have two corrugated furnaces, 3 feet 6 inches internal diameter. The total grate surface will be 168, and the heating surface, 5250 square feet. When the main engine is developing about 2,400 I.H.P., it is expected that each vessel will make a

speed of 17 knots. Upon the completion of these two boats, the Revenue Cutter Service will be represented on the Great Lakes by three fast vessels of the very best construction, to take the places of the antiquated sidewheelers *Johnson* and *Fessenden*, which have for years performed valuable service on those waters.

AUSTRIA.

Magnet.—The following additional particulars of the trial of this torpedo cruiser, described on page 805 of Volume VIII, are from the "Marine Rundschau," Berlin.

The working pressure in the four water tube boilers is 200 pounds per square inch.

The number of revolutions on the trial was from 280 to 282 per minute; vacuum, 24 inches; coal per I.H.P. per hour, 2.23 pounds. The total I.H.P., from one set of cards, was 5,946, distributed as follows. Starboard engine, H.P. cylinder, 878, I.P., 965, L.P., 1,115, collective, 2,958; port engine, H.P. cylinder, 876, I.P., 1,027, L.P., 1,085, collective, 2,988.

When running at one-half power, the coal consumption was 1.79 pounds per I.H.P. per hour.

Jaguar.—This torpedo cruiser, which was known by its building letter *A*, was launched in February. The following particulars are from the "Rivista Marittima," Rome.

The length is 301.8 feet; beam, 39.37 feet; mean draught, 14.1 feet; displacement, 2,264 tons. The hull is of steel and sheathed. The armament will consist of eight 4.7-inch and ten 1-85-inch rapid fire guns, two machine guns and two torpedo tubes.

The designed I.H.P. is about 7,700, and the expected speed, 20 knots. The boilers will be of the Dürr water tube type.

Another similar cruiser, building letter *B*, is projected.

BRAZIL.

Marshal Deodoro and **Marshal Floriano.**—Some particulars of these coast defense vessels were given on page 141 of the February number. The following additional data are from the "London Times."

The length will be 267.5 feet; beam, 48 feet; maximum draught, 13 feet 2 inches. There will be an armor belt of nickel steel 13.7 inches thick, and a protective deck 1.3 inches thick.

The armament will consist of two 9.45-inch guns in turrets, armored with 7.8 and 8.2 inches of nickel steel; four 4.7-inch rapid fire guns in casemates, armored with 2.8 inches of special steel; two 3.94-inch (?), four 6 and two 1-pounder rapid fire guns; two machine guns and two submerged torpedo tubes.

Lagrafel-d'Allest boilers will be used. The coal supply will be about 236 tons, and the complement, 200.

Timbira.—The third and last of the torpedo gunboats built by the Germania Co., Kiel, Germany, has arrived in Brazil. The sister vessel *Tupy* was described on page 142 of the February number.

CHILI.

Congreso.—This cruiser, or commerce destroyer, is building by the Forges et Chantiers de la Méditerranée at La Seyne, and will have, according to "Le Yacht," Paris, a displacement of about 7,381 tons, engines of about 15,750 I.H.P., and a speed of 23 knots. The protective deck will be of Harveyized steel, 1.57 inches thick, re-enforced in places to 4.33 inches.

The armament will consist of four 5.9-inch Canet rapid fire guns, protected by 4.92 inches of armor; eight 4.7-inch rapid fire guns and twenty-six smaller guns. The normal coal supply will be about 980 tons.

Injeniero Hyatt and **Cirujano Videla.**—These first class torpedo boats are two of the six building by Messrs. Yarrow & Co., and described on page 611 of Volume VIII. The *Hyatt* completed her official trial on February 15, the mean speed over the Maplin mile being 27.2 knots, with a steam pressure of 195 pounds per square inch.

ENGLAND.

Jupiter.—This first class battleship of the *Majestic* class, which was described on pages 161 and 167, Volume VII, and page 163 of Volume VIII, has completed satisfactory official trials.

To complete the description already given, it may be noted

that the eight single-ended boilers are each 16.25 feet in diameter and 10 feet 1.5 inches long, and have four furnaces. The total grate surface is 820, and the heating surface, 24,944 square feet. The following data of the trials are from "Engineering," London.

The four hours' forced draft trial was made on March 2. Draught, forward, 25.5 feet, aft, 26 feet 2½ inches; steam pressure, 150 pounds per square inch; vacuum, 26.6 inches; revolutions per minute, starboard, 102.4, port, 101.7; I.H.P., starboard, 6,348, port, 6,127, total, 12,475; speed, 18.4 knots; air pressure, .62 inch of water.

On March 3, the eight hours' natural draft trial was made. Steam pressure, 148 pounds per square inch; revolutions per minute, starboard, 93.9, port, 94.6; I.H.P., starboard, 5,047, port, 5,201, total, 10,248; speed, by log, 15.8 knots; air pressure, .2 inch of water.

The thirty hours' coal consumption trial was finished on March 5. Draught, forward, 24.5, aft, 24.75 feet; steam pressure, 146 pounds per square inch; vacuum, 26.6 inches; revolutions per minute, starboard, 81.5, port, 81.1; I.H.P., starboard, 3,116, port, 3,077; total, 6,193; coal per I.H.P. per hour, 1.51 pounds; speed, 14.1 knots.

The starting, turning and anchor trials were satisfactory.

Mars.—This first class battleship, described on page 386 of Volume VIII, has successfully completed her official trials in the English Channel.

The thirty hours' coal consumption trial was completed on March 16. The sea was smooth at the beginning, but the wind increased to half a gale during the trial. Draught at beginning, forward, 24 feet 2 inches, aft, 26 feet; steam pressure in boilers, 139.4 pounds per square inch; vacuum, starboard, 26.4, port, 26.3 inches; revolutions per minute, starboard, 83.74, port, 83.44; I.H.P., 6,028; coal per I.H.P. per hour, 1.94 pounds; speed, by patent log, 14.4 knots.

On March 18, the eight hours' natural draft trial was made. Draught, forward, 24, aft, 26 feet; steam pressure in boilers, 149.8 pounds per square inch; vacuum, starboard, 26.2, port, 26.6 inches; revolutions per minute, starboard, 97.57, port, 98.4;

I.H.P., 10,159; speed, by patent log, 15.96 knots; air pressure, .21 inch of water.

The four hours' forced draft trial was made on March 19, the draught being the same as on the previous day. Steam pressure, 146.5 pounds per square inch; vacuum, starboard, 26.5, port, 26.7 inches; revolutions per minute, starboard, 103.9, port, 104.5; I.H.P., 12,434; speed, by patent log, 17.7 knots; air pressure, .78 inch of water.

Canopus, Glory and **Ocean.**—The first two of these first class battle ships, described on page 807 of Volume VIII, were officially commenced on January 4, and the last one, on February 15.

Regarding this official date of beginning work on the *Ocean*, the "London Times" says:

"In anticipation of Monday, February 15, the date officially fixed for the commencement of the *Ocean*, which is to be built at Devonport, considerable work has already been done on the vessel. The keel plate has been laid on temporary blocks, and on Monday it will be lowered a few inches on to a permanent structure, which will carry the weight of the vessel throughout her construction. About 200 tons of material have already been built into her, and this quantity will be considerably increased by Monday. There is no precedent in the history of Devonport Dockyard for a vessel's being so far advanced before the date from which building is officially recognized."

The "Naval and Military Record" says the amount of material built into the hull was 250 tons, and that the first plate was on the blocks three weeks before February 15. It also adds that, at the end of March 13, when the *Canopus* and *Glory* had been building ten weeks, and the *Ocean*, six weeks, the amount of material built into the hulls was 1,300, 1,100 and 1,281 tons, respectively.

The side armor of the *Ocean*, extending about 250 feet on each side, will be of nickel steel, specially hardened. This armor will be about 18 feet in depth, extending from the protective deck, about 5 feet below the water line, to the main deck, about 13 feet above the water line. The protective deck, which is of concave shape, will be 2.5 inches thick, of two thicknesses of specially

hardened steel (not nickel). The cast steel stem weighs over 30 tons.

Besides the four 12-inch guns, which will be mounted in two hooded barbettes, and the twelve 6-inch rapid fire guns, which will be behind armored casemates, 5 inches thick, there will be twelve 12-pounder, six 3-pounder Hotchkiss rapid fire guns, eight .45-inch Maxim machine guns, and four 18-inch torpedo tubes, under water. No above water torpedo tube will be put into any of these vessels.

Niobe.—This first class sheathed cruiser, sister vessel of the *Diadem*, described on page 828 of volume VIII, was successfully launched on February 20 by the Naval Construction and Armaments Co., Barrow-in-Furness. The following description of the launching and the additional particulars of the hull and machinery are from "Engineering," London.

The fine lines of the vessel, combined with the excessive weight in proportion to the length, made special precautions necessary in the process of launching. The fineness of the lines, particularly forward, made it difficult, with the ordinary form of cradle, to give a good support to the forward end. The poppets or stilts of the forward cradle, instead of being bound together under the keel by chains, as is usual with merchant ships, were held together by steel plates which formed straps taking the form of the ship's section. These straps, extending for fifty feet of the length of the ship, were carried up to the heads of the poppets and connected to two fore-and-aft horizontal girders or shelves which extended along the tops of the straps. The heads of the poppets fit in under these shelves, so that there is little chance of their moving upwards or outwards. These girders at the fore end were turned down, to prevent the poppets from working out forward, and were stiffened by brackets, to prevent them from buckling under the pressure when the stern lifted. To make doubly sure, some of the chains used of old in bracing the two sides of the cradle were retained, and the poppets tied fore and aft by heavy ribands.

Amidships, between the hull and the top of the sliding ways, the packing consisted of solid wood laid fore and aft. Aft, where

the poppets had much less work to do, the heads of the poppets were bound with the usual cross chains and fore-and-aft ribands, the swelling of the ship in the way of the stern shafting being utilized to give a flat surface to take the heads of the poppets.

At the stern, where the ship was too fine for a cradle, the ship was supported by logs, stiffened with heavy steel channels, which were stretched across between and upon the ways.

The fineness of the ship minimized the length of the cradle. Sixty-eight feet forward and twenty-four feet aft were unsupported, so that the launching weight of 6,300 tons was carried on about 340 feet of the length, and to prevent any injury to the structure, particularly in the wake of the forward cradle, a very complete system of wooden pillars and horizontal and diagonal struts was put in between the protective deck and the shell plating. The spaces between the inner bottom were also chocked with timber over the ways. The necessity for the strong tying of the poppets on the sides of the ship at the fore end will be understood from the fact that when the stern became water-borne, which took place when the stern of the vessel was about 300 feet off the ways, there was a pressure of about 1,500 tons on the fore end of the ways.

The ways, which had to be exceptionally broad to reduce the pressure per square foot on the sliding surface, 1.84 tons in this case, were each 5 feet broad, and were laid at a declivity beginning at $\frac{13}{32}$ inch to the foot, increasing to $\frac{5}{8}$ inch under the stern and finishing in the water at $\frac{25}{32}$ inch per foot, the camber on the whole length of the standing ways being 2 feet 8 inches. This camber is given to the ways so as to keep their outer ends as low as possible without unduly increasing the initial declivity. The camber is, perhaps, greater than is adopted in other works, but this is necessitated by the fact that the building ground is rather more above high water mark than is usual. At Barrow they have a possible run of 250 yards. The time taken in launching at Barrow, from the first movement until the vessel is afloat, was, with the above run, for the second class cruisers of 360 feet, from 32 to 36 seconds, for the *Powerful*, 66 seconds, and for the *Niobe*,

58 seconds.* It may be mentioned that for destroyers, the Barrow firm, instead of laying the ways with a low initial declivity, increasing as the water is approached, reduces the final declivity by laying the ways with about one foot of a hollow, to prevent an excessive dip at the stern and thus possibly strain the light structure. These light vessels thus glide into the water almost on an even keel, the high initial declivity insuring an easy start. A 200-foot destroyer would travel 340 feet, going at an average speed of about 8.5 feet per second.

In accordance with the Admiralty practice, the deflection of the hull when water-borne was measured, and it was found that the *Niobe*, when afloat, had hogged, or dropped at the ends, $\frac{3}{4}$ inch. This amount, which was measured on a length of about 300 feet, was satisfactorily small. The opening and closing of the expansion joints in the boat deck and plating above the upper deck were automatically measured and recorded during the launching. The diagram obtained showed that there was little strain until the stern was in the water. When the stern was water-borne the vessel sagged, the opening in the deck closing to the extent of about .2 inch. The gap then opened to the extent of .21 inch from the normal. The gap again closed and opened, settling when the vessel was fully afloat with an opening .15 inch from the original position, due to the amount of the hog. The dip of the stern was observed from a boat, and making due allowance for the wave that rose under the counter, it was found that the heel of the vessel dipped to 32 feet under the general surface of the water.

The load displacement of the *Niobe* is 11,070 tons. The protective deck, armored casemates, conning tower and ammunition hoists weigh about 1,400 tons; the guns, with their shields, ammunition, stores, &c., weigh about 700 tons; the phosphor bronze stem, with its ram projecting 5 feet, weighs 16 tons; the two stern brackets weigh 26 tons; and the rudder, of cast bronze plated with brass, and 340 square feet in area, weighs 17 tons.

* The *Niobe* had four drags on each side, weighing in all 400 tons, including the connecting cables.

There are about 160 compartments above, and about 150 compartments below the protective deck.

The twin screw, triple expansion engines are to develop 16,500 I.H.P. at 110 revolutions per minute, with a steam pressure of 250 pounds per square inch at the engines. The crank shafts are in two parts of forged steel, and are 17½ inches in diameter, with a 9-inch hole from end to end. Each cylinder is supported upon two forged steel pillars at the front, and a strong cast steel column at the back, with cast iron guide plates attached for the piston rod crosshead. The main bearing frames, to which the columns are secured at their lower ends, are of cast steel, and are tied together longitudinally by cast steel distance pieces; the main bearings being of gun metal lined with white metal. Each engine has one air pump, worked by means of links and levers from the H.P. crosshead. A small hand pump is fitted in each engine room, arranged so that it may be driven by the main engines at any time if required. There are two main condensers, each having a cooling surface of 8,750, and two auxiliary condensers, each having a cooling surface of 1,100 square feet. There are four centrifugal circulating pumps for the main condensers which may also be used for pumping out the bilges; the combined capacity of these pumps will be about 4,800 tons per hour. The auxiliary condensers have also centrifugal circulating and independent air pumps.

The fire rooms are all forward of the engine rooms, and comprise four separate water tight compartments, three of which contain eight boilers each, and the remaining one, six boilers. Originally the boilers ordered for this vessel and her sister ships were of the old type of Belleville boilers; but since then, the new type, described on page 166 of the February number, with economizer in the uptake, has been installed. Six of the 30 boilers are composed of seven elements of large tubes and six of small or economizer tubes; 20 have eight elements of large and six of small tubes, and 4 have nine elements of large and seven of small tubes. Each large and small element contains 14 tubes of solid-drawn mild steel, the large tubes being 4½ inches, and the economizer tubes, 2¾ inches in external diameter. The small

elements are placed above the large ones at the base of the smoke casing, the space between—about 5 feet—forming a combustion chamber.

As the ship was well advanced before the change was determined upon by the Board of Admiralty, the boilers had to be designed to suit the head room in the various compartments, hence the number of tubes in each element of the economizer, which in later vessels ranges from 18 to 20, had to be cut down to 14. Originally the *Niobe* was to have 45,915 square feet of heating surface, now the economizers have 10,962, and the boilers proper, 29,588, together, 40,550 square feet. The ratio of the boiler to the economizer heating surface is 2.7 to 1. The total grate surface is 1,449 square feet.

Besides the auxiliary machinery already mentioned, there are in the engine rooms, two electric light engines and dynamos, two steering engines, two reversing engines, two turning engines, two hot well pumps, four fire and bilge pumps, two sets of evaporators and distillers, and two ventilating fans and engines. In the fire rooms are eight Weir's feed pumps, eight air blowing engines for supplying the furnaces with small jets of air under a pressure of about 30 pounds per square inch, eight forced draft fans with independent open engines, and eight double cylinder ash hoist engines. Two ship's ventilating fans are situated in the auxiliary machinery room aft of the main engine rooms, a third dynamo being placed in this room. The air compressors are placed in the capstan engine room, and the air reservoirs in the submerged torpedo room forward. The whole of the main and auxiliary machinery will combine upwards of 124 steam cylinders.

The main steam and receiver pipes are of lap-welded steel, having a butt strap riveted over the weld, and are fitted at various positions with gun metal expansion joints. There are in all four main steam pipes between the boilers and the engine rooms, each fitted with Belleville steam separators. There are four smoke pipes placed in line fore and aft, the height being about 80 feet above the grates. The diameter of the forward pipe is 7.5, of the second, 8 feet, and of the two after ones, 8.25 feet.

Europa.—This first class sheathed cruiser, a sister vessel to the *Diadem*, was successfully launched on March 20 by the Clydebank Shipbuilding and Engineering Co., the new name of the old firm of Messrs. J. and G. Thomson.

Doris.—This sheathed second class cruiser completed her official trials in March. The results of the natural and forced draft trials were given on page 162 of the February number. The 30 hours' coal consumption trial resulted as follows.

Steam pressure in boilers, 143 pounds, at engines, 140 pounds per square inch; vacuum, starboard, 26.5, port, 26.8 inches; revolutions per minute, starboard and port, 118.7; I.H.P., starboard, 2,463, port, 2,475; total, 4,938; air pressure, *nil;* speed, 16.5 knots; coal per I.H.P. per hour, 1.47 pounds.

Isis.—This sheathed second class cruiser, a sister vessel of the *Dido*, described on page 388 of Volume VIII, was built by the London and Glasgow Engineering and Shipbuilding Co., Govan, Scotland, launched on June 27, 1896, and finished the official trials on February 25. The principal data of the vessel are as follows.

Length between perpendiculars, 350 feet; breadth, extreme, 54 feet; mean load draught, 20 feet 6 inches; displacement at this draught, 5,600 tons. The hull is built of Siemens-Martin steel, and is sheathed with 3½ inches of teak and copper. Bilge keels of teak, covered by naval brass plates, are fitted for about 156 feet amidships. The armament consists of five 6-inch, six 4.7 inch, nine 12-pounder and seven 3-pounder rapid fire guns, and a number of smaller machine guns; one torpedo tube with 3-inch steel mantlet, discharging above water aft, and two submerged tubes on broadside forward, all fitted for 18-inch torpedoes.

The two sets of inverted triple expansion engines, supplied by the builders of the hull, have cylinders 33, 49 and 74 inches in diameter, with a stroke of 39 inches. There are two boiler rooms, each containing four single-ended boilers, each boiler having three corrugated furnaces. The working pressure is 155 pounds. The designed I.H.P. and speed were, for natural draft, 8,000 and 18.5 knots, and for forced draft, 9,600 and 20 knots, the latter power to be obtained with an air pressure not greater

than 1.5 inches of water. The weight of the boilers is 549 tons, and of the whole machinery, 900 tons.

On February 19, the eight hours' natural draft trial was made with the following results: Draught, forward, 19 feet 10 inches; aft, 21 feet 8 inches ; steam pressure, 149 pounds per square inch ; vacuum, starboard, 27.7, port, 28.5 inches; revolutions per minute, starboard, 134.6, port, 137.7 ; I.H.P., starboard, 4,121, port, 4,087, total, 8,208 ; air pressure, .41 inch of water ; speed, by log, 19.8 knots.

The four hours' forced draft trial was made on February 20. The mean draught was 20.5 feet; steam pressure, 150 pounds per square inch ; vacuum, both condensers, 27.9 inches ; revolutions per minute, starboard, 143.4, port, 146.5 ; I.H.P., starboard, H.P. cylinder, 1,635, I.P., 1,664, L.P., 1,615, collective, 4,914; I.H.P., port, H.P. cylinder, 1,614, I.P., 1,615, L.P., 1,697, collective, 4,926; I.H.P., total, 9,840; air pressure, 1 inch ; speed, by patent log, 21.2 knots.

The highest power developed was 4,958 I.H.P., starboard, with 143.4, and 5,168 I.H.P., port, with 145 revolutions per minute. The minimum was 4,874 I.H.P., starboard, with 142.7, and 4,730 I.H.P., port, with 143 revolutions per minute.

On February 23, the thirty hours' coal consumption trial was completed. Draught, forward, 19 feet 4 inches, aft, 21 feet 3 inches; steam pressure, 145 pounds per square inch; vacuum, starboard, 26.8, port, 26.5 inches; revolutions per minute, starboard, 116.5, port, 119.2; I.H.P., starboard, 2,441, port, 2,484; total, 4,925 ; speed, by log, 17.5 knots ; coal per I.H.P. per hour, 1.6 pounds.

During all the trials the machinery worked well. From the data of the full power trial and the weights, it will be found that 17.92 I.H.P. were developed for one ton of boiler, and 10.93 I.H.P., per ton of all machinery.

Hermes, Highflyer and **Hyacinth.**—These second class cruisers, which were described on page 165 of the February number, are to have a battery different from the one there given. The proposed battery will consist of eleven 6-inch, nine 12 and six 3-pounder rapid fire guns.

Of the 870 tons, given as the weight of the machinery, 470 tons are for the boilers. As will be seen from the weights given tons above for the *Isis* and *Dido*, the newer cruisers will have 79 less of boiler weights and 30 tons less weight of all machinery.

Pelorus.—The following additional description of the hull and machinery of this third class cruiser, the trials of which were given on page 166 of the February number, is from "Engineering," London.

The protective deck is connected with the framing and plating of the side of the ship 2 feet 6 inches below the water line, and in the center rises 1 foot 3 inches above it. It is 2 inches thick on the slopes and in the wake of the machinery, where the rise of arch is increased to insure that the cylinders will be completely under the deck. There is a double line of coal bunkers on either side of the boiler casings above the protective deck, and a single line below. The shell plating is 12½ pounds, and bilge keels, 18 inches deep, are fitted. The boiler compartments occupy 80 feet of the length of the ship, being 24 feet wide, while the engine room is 38 feet long and the full width of the ship.

The cylinders are independent castings, and the L.P. cylinders only are jacketed with steam. They are supported upon wrought steel columns strongly braced together. The valves of the H.P. and I.P. cylinders are of the piston type, while the L.P. cylinders are fitted with flat double-ported slide valves. Balance cylinders are fitted for the purpose of reducing the strain on the valve gear, which is of the double eccentric link motion type, reversing by a double cylinder engine and gear of the "all-round" type. The air pumps are driven from the crossheads of the L.P. engines.

The condensers are of brasss, and are placed in the wings of the vessel, the water being circulated through each condenser by a 13-inch Gwynne centrifugal pump. The crank and propeller shafts are of Vickers' steel. The thrust blocks and collars are of cast steel; the latter are lined with white metal and are of the horseshoe type, each separately adjustable. The screw propellers are three-bladed, with bosses of gun metal and blades of

Parsons' manganese bronze. The diameter of the propellers is 11 feet, and the pitch, 11 feet. In the main engine rooms there are two fire and bilge pumps, two hotwell pumps, two evaporators and two distillers with pumps, &c., complete, two sets of engines and dynamos, a set of air compressing engines and pumps, and the steering engine.

The working pressure in the Normand boilers is 300 pounds, this being reduced to 250 pounds per square inch at the engines. The boiler tubes are of solid-drawn steel, and are galvanized externally in an electro-depositing bath. The stokeholds are arranged so that the boilers can be worked under natural as well as forced draft. Air for the latter is supplied by eight fans, each 5.5 feet in diameter, and driven by an independent engine of the open type, as generally supplied by machinery contractors.

Pegasus.—This third class cruiser, building by the Palmer Shipbuilding & Iron Co., Newcastle-on-Tyne, and described on page 391 of Volume VIII, was successfully launched on March 4. The following additional particulars are from "Engineering," London.

The hull is built of steel. A sloping protective deck extends the whole length of the vessel. It extends 2 feet 6 inches below the load water line at the sides, and is formed of two thicknesses of plating in the wake of the machinery; the sloping portion is 2 inches, and the horizontal part, 1 inch thick. The engine rooms are separated by a longitudinal water tight bulkhead. Throughout the boiler spaces longitudinal water tight bulkheads extend at the sides and form coal bunkers. Above the protective deck, and extending the length of the machinery spaces, there is a double line of coal bunkers. The conning tower is placed on the forecastle deck, and immediately above is built the navigating bridge and search light platform.

The armament consists of eight 4-inch and eight 3-pounder rapid fire guns in broadside, and three .45-inch Maxim guns, and two 14-inch torpedo tubes. Two of the 4-inch guns are on the forecastle, two on the poop, and four on the upper deck between the poop and the forecastle. The 3-pounders are on the upper deck. Two of the .45-inch Maxim guns are on the hammock

berthing amidships, and one on the forecastle deck. The torpedo tubes are on the upper deck, one on each broadside.

The twin screw, vertical triple expansion engines, capable of developing 7,000 I.H.P. with a pressure of 250 pounds per square inch, have cylinders 20.5, 33 and 54 inches in diameter, the stroke being 27 inches. Steam is supplied by eight of Reed's water tube boilers, with a working pressure of 300 pounds per square inch, which is reduced to 250 pounds at the engines. All auxiliary machinery is placed below the protective deck in the machinery spaces. Two fans, 5.5 feet in diameter, are fitted in each fire room for the supply of air. All water tight doors are worked from the upper deck.

The installation of electric lights includes 350 incandescent lamps, two search lights of 25,000 candle power and two yard-arm reflectors. The dynamos and engines are placed under the protective deck in the engine room.

The vessel has two masts with a light fore-and-aft rig, and two smoke pipes. The complement will consist of 224 officers and men.

Star.—This 30-knot torpedo boat destroyer, built by the Palmer Shipbuilding and Iron Co., Jarrow-on-Tyne, and described on page 841 of Volume VIII, started on her official trial from Portsmouth on April 5, but did not complete it, owing to leaky piston rod glands. A new patent metallic rod packing had been used. The contract required that the vessel should maintain a speed of 30 knots for three hours, the engines to develop 6,000 I.H.P. at about 420 revolutions per minute, and the coal consumption not to exceed 2.5 pounds per I.H.P. per hour. If the coal consumption exceeded this amount, an extra weight of 10 tons for every $\frac{1}{10}$ pound of coal in excess was to be added to the new trial weight.

The vessel started on the trial with a displacement of 319 tons, and ran well, without appreciable vibration, for 2 hours and 10 minutes, when the trial was discontinued. For this time, the contract I.H.P. was exceeded, although the number of revolutions per minute did not exceed 395. The speed for the entire time was 30.05 knots, and for the six runs over the measured mile,

30.3 knots. The first, third and fifth runs over the mile were with the wind and against the tide, and in the other three runs the conditions were reversed. The miles were run, in order, at the following speeds: 29.5, 29.5, 30.1, 30.3, 30.83 and 31.+, the last mile being run in 1 minute and 54 seconds, giving a speed slightly over 31 knots.

Had not the trial been stopped by the leaky glands, a serious accident might have resulted, as an examination, made after the vessel was docked, showed that the hubs of both propellers were badly cracked.

The following information is from the "London Times."

New propeller hubs having been fitted, another three hours' trial was made on April 22. Indicator cards were taken every 20 minutes. The speed for the six runs over the mile was 30.03 knots, the revolutions being 386 per minute; for the three hours, the figures were 30.103 knots and 389 revolutions. The maximum speed attained was 31.48 knots. When the calculations had been made, it was found that the coal consumption was very slightly above the 2.5 pounds. Another trial was, therefore, commenced on April 29, the trial weight being increased by 2 tons. After running at 30 knots for about 15 minutes, the S.L.P. cylinder split, and steam rushed into the engine room. Although the boiler stop valves were promptly closed, five men were seriously scalded. The vessel returned to Portsmouth, using her port engine and being helped by a tug.

The four Reed boilers are in two compartments, with two blowers in each compartment.

On the run from the contractors' yard to Portsmouth, the vessel is said to have made 32.25 knots for a short time.

Fame.—This 30-knot torpedo boat destroyer, built by Messrs. J. I. Thornycroft & Co., Chiswick, has successfully completed her trials. The previous trials were mentioned on page 173 of the February number.

The displacement on a draught aft of 7 feet 2 inches is given as 272 tons, and the designed I.H.P., as 5,400. On the full power trial, made about February 22, the draught forward was 5 feet 1½ inches, and aft 7 feet 1 inch. The following results

were obtained. Steam pressure, 205 pounds per square inch; vacuum, starboard, 26.1, port, 26.4 inches; revolutions per minute, starboard, 394.8, port, 393.1; I.H.P., starboard, 2,914; port, 2,980, total, 5,894; air pressure, 4 inches of water; speed, 30.155 knots.

On April 15, another trial was made off Chatham dockyard, the mean speed for the six runs over the mile being 30.021, and for the three hours, 30.168 knots.

Angler and **Ariel.**—These 30-knot torpedo boat destroyers, built by Messrs. J. I. Thornycroft & Co., were successfully launched on February 2 and March 5, respectively.

Albatross.—The following particulars are given of this 32-knot destroyer, building by Messrs. J. I. Thornycroft & Co.

Length, 227 feet; beam, 21.25 feet; draught, 8 feet 4 inches; diaplacement, 360 tons; I.H.P., 7,500.

The armament will consist of one 2.99-inch and five 2.24-inch rapid fire guns and two deck torpedo tubes.

Whiting.—This 30-knot torpedo boat destroyer, built by the Palmer Shipbuilding and Iron Co., and described on page 841 of Volume VIII, has made preliminary contractors' trials. She was run a number of times over the measured mile at an average speed of 30.2 knots, and was kept running for three hours at an average speed of 30.1 knots.

Bat.—Another 30-knot destroyer, built by the Palmer Shipbuilding and Iron Co., was taken out to sea on March 8 for the full speed trial. The conditions were not favorable for a high speed, there being a heavy swell on, which caused the vessel to pitch and roll. Six runs over the measured mile gave an average speed of 30.7 knots. The highest speed with and against the tide during the mile runs was 31.2 knots. The average speed for the three hours was 30 knots. The engines ran for a considerable part of the time at over 400 revolutions per minute, sometimes as high as 416. The engines and boilers worked in a highly satisfactory manner in spite of the unfavorable conditions.—"Engineer's Gazette," London.

Fawn and **Flying Fish.**—Sister destroyers of the above, and built at the same yard, were launched on April 13 and March 4, respectively.

Seal.—This 30-knot destroyer, built by Messrs. Laird Bros., Birkenhead, was launched on March 6.

Gypsy.—This 30-knot destroyer, built by the Fairfield Co., Govan, was launched on March 9.

Bittern and **Leopard.**—These 30-knot destroyers, built by the Naval Construction and Armaments Co., Barrow, were launched on February 1 and March 20, respectively.

FRANCE.

Jauréguiberry.—This battleship finished her contract trials on February 12. The following data of the trials are from "Le Yacht," Paris.

On January 20, the repairs to the boilers made necessary by the accident in June (see page 624 of Volume VIII) were completed, and a preliminary trial was attempted, but was interrupted by defects in some of the steam drums or collectors. During the repairs, solid-drawn tubes were put in place of the welded ones directly exposed to the fire, as well as in the two rows of tubes around the sides of the special fire boxes, the latter being a special feature of this type of boiler. All the old furnace and ashpit doors were replaced by doors which close automatically.

On January 26, the official, six hours' natural draft trial at 3,450 I.H.P. was made. During this time, six runs were made over the measured mile, the speed being probably reduced by the heavy swell. The mean results were: Revolutions per minute, 61.7; speed, 12.25 knots; I.H.P., 3,386; coal per I.HP. per hour, 1.54 pounds, or less than the contract requirement.

On February 2, a six hours' coal consumption trial at 7,890 I.H.P. was made, the vessel being practically at her load draught of 26.57 feet. With 142 pounds of steam, the number of revolutions per minute was about 83, giving a speed of 15.8 knots for the six runs over the mile. Although the weather was fine, the speed was slightly reduced by the swell. The I.H.P. was about 7,965. The coal per I.H.P. per hour, 1.51 pounds, again less than the contract requirements.

On February 11 to 12, a twenty-four hours' full power trial was satisfactorily made, the I.H.P. for two hours being 13,315,

and the number of revolutions per minute, about 95. The speed obtained by running over the mile was 17.4 knots. For eighteen hours of this trial the coal was measured, the average I.H.P. being 10,208 with 88.5 revolutions per minute, and the coal per I.H.P., 1.57 pounds her hour. The lowest contract limit was 1.79 pounds per I.H.P. per hour.

The total coal burned during the 24 hours was 419,205 pounds, or about 7.8 tons per hour. The speed during this time was estimated at slightly more than 17 knots. With the normal coal supply of about 1,000 tons, the radius of action at 17 knots would be 2,179 nautical miles.

For purposes of comparison and calculation, the principal data of the machinery are here repeated, and the results of the various trials tabulated.

The twin screw, triple expansion engines have cylinders 43.3, 64.2 and 97 inches in diameter, the stroke being 44 inches. They were designed for 13,000 I.H.P. at 97 revolutions per minute, with a steam pressure of 170 pounds per square inch at the engines. The total cooling surface is 17,860 square feet. The 24 Lagrafel-d'Allest boilers are arranged in six fire rooms and have a total grate surface of 1,085, and a heating surface of 35,092 square feet. The working pressure in the boilers is 213 pounds per square inch. The weight of boilers, water, pumps, floors, pipes, &c., is 551.7 tons, and that of all the machinery, 1,120 tons.

Date of trial	May 20, '96	May 27, '96	Feb. 11-12,'97	Feb. 2,'97	Jan. 26,'97
Duration, hours	6	4	18	6	6
Speed, knots	17.6	17.67	17.*	15.8*	12.25
Revols. per minute	96	...	88.5	83	61.7
I.H.P.	13,630	13,690	10,208	7,965*	3,386
Coal per sq. ft. of grate surface per hour	22 23	31.46	14.77	11.09	...
Coal per I.H.P. per hour	1.77*	1.87	1.57	1.51	1.54
No. boilers in use	24	18	24	24	...

*About.

Charles Martel.—This battleship, the preliminary trials of which were given on page 847 of Volume VIII, has had the welded tubes of her Lagrafel-d'Allest boilers renewed, as a result

of the accident on the *Jauréguiberry* (see page 624 of Volume VIII). The new tubes are solid-drawn, and are .137 inch thick, the old welded tubes being .117 inch thick. The old furnace and ash pit doors have been replaced by doors which will close automatically when a leak occurs inside the furnace.

Masséna.—This triple screw battleship, built by the Chantiers de la Loire, St. Nazaire, was partially described on page 603 of Volume VII. The following additional particulars are from "Le Yacht," Paris.

Bilge keels are fitted. The protective deck is flat and made of two courses of plates, and is on a level with the upper edge of the belt armor. Parallel to this deck, and on a level with the lower edge of the belt armor, is a splinter deck, 1.57 inches thick. Above the belt armor, the side armor is 3.94 inches thick, coffer dams being fitted behind it. The conning tower is 13.8 inches thick.

There are only four torpedo tubes, two above and two under water, for 17.72-inch torpedoes.

The three engines are of the vertical, four cylinder, triple expansion type with cylinders 35.04, 52 and (2) 60.6 inches in diameter, the stroke being 33.07 inches. They were designed to develop 9,074 I.H.P. with 106 revolutions, and 13,315 I.H.P. with 120 revolutions per minute under forced draft, and burning 2.23 pounds of coal per I.H.P. per hour. Under natural draft, running at 116 revolutions per minute, and burning between 1.97 and 2.19 pounds per I.H.P. per hour, the expected I.H.P. is 11,836, and the speed, 17 knots.

The pistons are of cast steel and the cylindrical condensers of brass, with brass tubes. Marshall's valve gear is used. The engines were built by the same company at their St. Denis yard.

The twenty-four Lagrafel-d'Allest boilers are arranged in three groups and have one furnace each. The working pressure is 213 pounds per square inch. Total grate surface, 1,076.4, and heating surface, 32,292 square feet; ratio of G.S. to H.S., 1 to 30. The total weight of boilers is about 439 tons. With forced draft, 31 pounds of coal may be burned per square foot of grate surface. The two smoke pipes are each 65.6 feet above the grates.

There are four dynamos of 600 amperes at 83 volts, three of which are sufficient for the six search lights, electric motors and other lights.

Jena.—Orders for the construction of this triple screw battleship have been given to the dockyard at Brest. The vessel first proposed was referred to as A_3 on page 175 of the February number. The following dimensions now given by "Le Yacht," Paris, show the ordered vessel to be more like a larger *St. Louis* or *Gaulois* than a *Henri IV.*

The proposed length is 400.79 feet; beam at the water line, 68.24 feet; beam, maximum, 68.37 feet; load displacement, 11,862 tons. The I.H.P. under forced draft is to be 15,288, and the corresponding speed of the vessel, 18 knots. The normal fuel supply will be about 800 tons, consisting of coal and liquid fuel; this can be increased to about 1,085 tons. There will be a complete armor belt.

The armament will consist of four 12-inch guns, arranged in pairs in two turrets. In the armored citadel between the turrets will be placed eight 6.49-inch rapid fire guns. On the deck above these, there will be eight 3.94-inch rapid fire guns. Half of these sixteen rapid fire guns are arranged to fire abeam and astern, and the other half, abeam and ahead. The 6.49-inch guns will have shields 2.76 inches thick, and the 3.49-inch guns, shields 2.13 inches thick. There will be, in addition, sixteen 1.85-inch and eight 1.46-inch rapid fire guns.

The three engines will be of the vertical, inverted, triple expansion type. The twenty Belleville boilers will be fitted to burn coal and liquid fuel mixed.

Amiral Pothuau.—This armored cruiser, described on page 848 of Volume VIII, finished her trials on February 26. The following data are from "Le Yacht," Paris.

The full power, forced draft trial, was finished in the latter part of February off Cherbourg. The contract speed was 19 knots, with 9,863 I.H.P. The trial resulted in a speed of 19.2 knots, from runs over the measured base, and 10,256 I.H.P. The coal burned per square foot of grate surface was 24.17 pounds, and per I.H.P. per hour, 1.89 pounds. The normal coal capacity

is 530 tons, but this can be increased to 628 tons. As the coal consumption is about 207 tons per day, the steaming radius at a speed of 19.2 knots, and with 628 tons on board, would be 1,382 nautical miles.

On February 26, a 24-hours' trial was completed, the mean speed being 17.8 knots. This ended the official trials.

D'Assas.—This second class cruiser, described on page 404 of Volume VIII, is nearly ready for her trials. The following additional particulars are from "Le Yacht," Paris.

The contract weight of the twin screw engines, the d'Allest boilers and their accessories was 700.8 tons, and of the hull and armor, 1,745 tons. This weight has been exceeded by about 100 tons. The armament has not yet been installed. There are three smoke pipes and two short masts.

On the trip from the builders' works at St. Nazaire to Brest, the engines were run at 60 revolutions per minute, giving a speed of about 10 knots.

The speed expected with natural draft is 18 knots.

Fleurus.—Further trials of this third class cruiser were begun at Cherbourg on January 20, but were interrupted by faulty setting of the boiler tubes. The boilers of this vessel are of the French Admiralty type and are new, having been put in during 1896, as mentioned on page 626 of Volume VIII. "Le Yacht," Paris, in its issue of March 2, 1897, says that, as the engines have always worked well during the numerous trials, they have been accepted. The new boilers are to be given another trial at reduced power and for a shorter period. A speed of 17 instead of 18 knots will also be considered satisfactory. On April 17, the same paper says that, owing to new signs of weakness manifested during the last trial of the the boilers, the latter will, probably, be renewed entirely.

GERMANY.

Victoria Luise.—This steel, triple screw, second class cruiser, described as cruiser *L* on page 828 of Volume VII, and in the table facing page 556 of Volume VIII, was successfully launched by the Weser Works, Bremen, on March 30.

The length over all is 359.25 feet, between perpendiculars, 344.5 feet; beam, 57.1 feet; draught, normal, 20.5 feet, load,

21.68 feet; displacement at normal and load draughts, 5.561 and 6,004 tons, respectively; I.H.P., natural draft, 8,877, forced draft, 9,863; speed, at normal displacement, 18.5; maximum speed, 20 to 21 knots. Coal supply at normal draught, 490 tons, and at load draught, 935 tons.

The protective deck is 1.56 inches thick on the horizontal part above water, and 3.94 inches on the inclined sides below water, the lower end being 5 feet below, and the upper horizontal part, 20 inches above the normal water line. On each side, for a length of about 230 feet, there are coffer dams, which are packed with cork and are about 8.25 feet high and 27.5 inches wide. The bases of the smoke pipes are protected by a glacis, 4.72 inches thick.

The armament will consist of two 8.27-inch, 40-caliber guns in turrets, 3.94 to 8.27 inches thick; four 5.9-inch rapid fire guns in turrets, 3.94 to 5.9 inches thick, and four in casemates of the same thickness; ten 3.46-inch rapid fire guns, protected by shields; ten 1.46-inch rapid fire guns; four machine guns; and three submerged torpedo tubes, one in the bow and one on each broadside.

There will be two military masts, the forward one having two, and the after one, a single fighting top.

There are three vertical, four cylinder, triple expansion engines, each engine being in a separate compartment. The Dürr water tube boilers are arranged in six separate compartments.

Herta.—A sister vessel to the above, and known as cruiser *K*, was successfully launched by the Vulcan Works, Stettin, on April 14. She also has three screws.

———, building letter *G*.—This twin screw, fourth class cruiser has been commenced at the Germania Co.'s works at Kiel. The principal data are as follows.

Length, 328.1 feet; beam, 38.7 feet; draught, 15.74 feet; displacement, about 2,560 tons.

The protective deck will be 1.97 inches thick. The armament will consist of eight or ten 4.13-inch rapid fire guns and twelve smaller guns.

The triple expansion engines are to develop about 5,918 I.H.P. and give the vessel a speed of 20 knots.

GREECE.

Psara.—This battleship, built by the Forges et Chantiers de la Méditerranée in 1890, has recently undergone some changes in her armament and in her masts. The sister vessels *Hydra* and *Spetsai*, also built in France, are to be changed in the same way. The latter vessel was briefly described on page 320 of Volume IV, some of the dimensions being given in metric units. The following particulars from "Le Yacht," Paris, will apply to all three vessels, so soon as the changes have been made in the *Hydra* and *Spetsai*.

The hull is of steel; the length over all, 334 feet; beam, 51.84 feet; depth of hold, 29.69 feet; displacement, 4,414 tons. There is a complete belt armor, 11.81 inches thick and 3.94 feet wide, its upper edge being approximately on the water line. Above this belt, the vessel's side is 1.64 inches thick. Forward there is a two-story citadel, and aft, a turret, both having a maximum thickness of 13.78 inches of Creusot armor. Between these two, and extending forward to the bow, there is a central superstructure. There are 118 watertight compartments. The normal coal capacity is about 600 tons.

The armament consists of three 10.63-inch, 36-caliber Canet guns, two in the upper story of the citadel forward, and the other in the turret aft; five 5.9-inch, 36-caliber Canet guns, one in the forward part of the upper story, two in the forward and two in after part of the lower story of the citadel; one 3.94-inch Canet gun in the superstructure at the bow; eight 2.56-inch Canet guns in the superstructure, two forward and six aft of the citadel; four 1.85-inch guns furnished by the Greek Government; twelve machine guns in the tops and forward on the top of the citadel; and three torpedo tubes, one forward and two in broadside. Besides these, there are still on board four 3.42-inch guns, which are to be removed.

The twin screw engines, of the horizontal triple expansion type, were designed to develop 6,608 I.H.P. under forced draft, and give the vessel a speed of 17 knots. There are four cylindrical double-ended boilers, the working pressure being 142 pounds per

square inch, and one auxiliary boiler. There are two oval smoke pipes.

Originally there were three masts fitted with fore-and-aft sails. The forward one has been cut down low and fitted with a large platform, on which are mounted four of the smaller guns. The main mast has been shortened about six feet, lowering the military top by that much. The mizzen mast has been cut down low and fitted with one military top and a large platform below.

The complement consists of about 400 officers and men.

JAPAN.

Fuji.—This battleship, previously described on page 419 of Volume VIII, and under the sister vessel *Yashima*, on page 414 of Volume VIII, has finished satisfactory official trials. The following additional particulars of the vessel are from "Engineering," London, and the trial data, from the "London Times" and other sources.

The contract was signed in London on June 7, 1894. The time for completion originally was to have been five years, to suit the financial arrangements of the Japanese Government, but the war with China breaking out led to the period being shortened to 33 months. This period has, however, been extended by the strike at Messrs. Armstrong & Co.'s works.

The height of muzzle of the four 12-inch guns is 25 feet above the load water line. The designed number of revolutions per minute was 120 when the machinery was developing 14,000 horse power, the air pressure in the stokehold not exceeding $1\frac{1}{2}$ inches of water. The revolutions would be 108 per minute when developing 10,000 horse power with open stokeholds. The ten single-ended cylindrical boilers were built for a working pressure of 155 pounds per square inch.

The vedette boats are each fitted to carry two fish torpedoes in dropping gear, and a $2\frac{1}{2}$-inch Hotchkiss gun in the bow.

There are two dynamo rooms, one forward and one aft, and there is also an electric light installation in the air compressing room aft. These electric light stations are on the lower deck.

The lighting installation consists of three dynamos and engines, each giving 400 amperes at 80 volts, the dynamos being of the direct current type, with self regulators. There are five search lights fitted with Mangin mirrors, 24 inches in diameter.

The air compressing plant, for charging the air chambers in the fish torpedoes, is in two rooms on the lower deck beneath the citadel. The plant is in duplicate, as are most other features in the ship, the sensible plan being followed of making one end of the citadel as far as possible independent of the other, so that one could be fought after the other had been put out of action. The two rooms contain four air compressing engines and three reservoirs of 300 tubes combined. The compressed air is led by pipes to the torpedo room on the deck below, there being two charging columns in each torpedo room and one for the bow discharge. The total capacity of the reservoirs is 66 cubic feet at 1,250 pounds maximum pressure.

The hydraulic machinery is placed in two rooms on the lower deck. The power is used for working the barbette turntables, hoisting ammunition, elevating and loading guns, &c. These rooms are respectively beneath the barbettes which they serve, and are contained within the citadel. They are all well ventilated, and have tiled floors in accordance with Japanese custom.

For ventilating all the compartments below the water line, there are four fans arranged in pairs, two forward and two aft below the protective deck and within the citadel. Sluice valves with floats are fitted in the air trunks at each watertight bulkhead.

Steering is effected from various positions in the vessel, either by the hydraulic telemotor or by small controlling wheels with shafting and gearing. For steering by hand, the usual steering wheels are placed in the engine compartment, the ordinary complement of men for the purpose being 16.

The ship's complement will consist of one admiral, 36 officers, 15 midshipmen, 17 warrant officers, 115 petty officers and 416 men.

On March 18, the six hours' natural draft trial was made in the vicinity of the Downs at the mouth of the Thames, where a

course of ten nautical miles had been laid off. Four runs were made over this course. The mean results were as follows.

Steam pressure, 150 pounds per square inch; vacuum, starboard, 27.75, port, 28.3 inches; revolutions per minute, starboard, 109.3, port, 108; I.H.P., 10,200; speed, 16.8 to 16.937 knots. Fire rooms open during trial.

On March 19, the full power, forced draft trial resulted as follows, six runs over the course being made.

Steam pressure, 152 pounds per square inch; vacuum, starboard, 28, port, 27.3 inches; revolutions per minute, starboard, 120.3, port, 120; I.H.P., 14,100; speed, 18.5 to 18.655 knots; air pressure, 1.5 inches of water. The vessel on this trial was at her deep load draught, having 734 tons of water and other ballast on board to cover deficiency in weight of armament, &c., not yet on board. The delay in supplying the armament is due to the strike at the Armstrong works.

Heeling experiments made with the *Fuji* showed that the actual metacentric height was 3.48 feet, instead of 3.53 feet, as calculated,—a very close agreement.

———.—The contract for the building of this battleship was let to the Thames Iron Works and Shipbuilding Co. on February 4. The following particulars are from "Engineering," London.

The length over all will be 438 feet, and between perpendiculars, 400 feet; beam, 75.5 feet; draught, 27.25 feet; displacement at above draught and with 700 tons of coal, 14,850 tons.

The armor is to be of Harveyized nickel steel and disposed as follows: the hull to have a belt from stem to stern, 9 inches thick all through engine, boiler and magazine spaces, and 8 feet 2 inches deep; tapering at the ends beyond from 7 to 4 inches in thickness. Above this an additional belt of 6-inch armor will be worked to the height of the main deck and for a length of 250 feet, inclosing the two barbettes. At each end of this space there will be a curved transverse bulkhead, 14 inches thick, extending from above the protective deck to the height of main deck, thus forming a complete citadel 250 feet long. Above the main deck, the two barbettes, which are circular in form and protected by 14-inch armor with 4-inch teak backing, rise to a height of 4 feet

above the upper deck. Between the main and upper decks, screen bulkheads will extend from the barbettes to the ship's side. The protective deck will rise from the lower edge of the belt armor to about 3 feet above the water line, and be 3 inches thick on the flat part and 5 inches on the sloping sides, tapering at the ends.

Each barbette will carry two 12-inch breechloading guns of 40 calibers. Fourteen 6-inch quick-firing guns of 40 calibers will be placed in casemates 6 inches thick. Eight casemates will be on the main deck and six on the upper deck, and are to be fitted with the usual dismounting and stowing gear. The casemates will be made watertight both on the inner and outer sides, thus protecting the men at the guns from any explosive shells that might enter between decks, and also preventing the admission of water between decks should the gun port get damaged. In addition, there will be twenty 12-pounders on the upper deck, eight 1.85-inch quick-firing guns on upper and main decks and in the military tops, and four 1.85-inch quick-firing guns on bridges and upper works. There will be five 18-inch torpedo tubes, one protected by armor in the stem above water, and four submerged. The usual torpedo net defence will be fitted.

The twin screw, triple expansion engines, to be built by Messrs. Humphreys, Tennant & Co., Deptford, are to develop 14,500 I.H.P. and give the vessel a speed of not less than 18 knots. The cylinders will be 34, 53 and 84 inches in diameter, the stroke being 48 inches. There will be 25 new type Belleville boilers, with a total heating surface of nearly 40,000 square feet. There will be three smoke pipes and two military masts.

The electric light installation will consist of four sets of combined engines and dynamos, the latter of the direct current type. Three sets will be of 400 amperes at 80 volts each, and one of 200 amperes at 80 volts. Six 24-inch search lights of 20,000 candle power each, and about 900 16-candle power incandescent lamps will be provided.

The steam steering gear will be on Cameron's self-regulating principle, worked by steam steering engines in duplicate, and fitted with Messrs. Brown's telemotor, controlling the helm from

the pilot house, the bridge forward, the after pilot house, and the bridge and steering compartment, and from below the protective deck forward.

There will be 14 boats, including two 56-foot vedette boats, fitted with the Thames Iron Works water tube boilers, one 42-foot launch, and one 30-foot steam pinnace. Each of these will carry Whitehead torpedoes, and be fitted for mining and countermining purposes. The vessel was designed by Mr. Mackrow, and is to be completed in twenty-seven months. The total complement of men and officers will be 741, including an admiral and 38 officers.

NORWAY.

Tordenskjold.—This coast defense vessel, a sister to the *Harald Haarfagre* (described on page 183 of the February number), was successfully launched on March 18 by the same builders, now known as Sir W. G. Armstrong, Whitworth & Co., Newcastle-on-Tyne.

The armor belt, 6.5 feet wide, extends about 170 feet of the length of the vessel, 7-inch transverse bulkheads being built at the ends. The sloping protective deck is 2 inches thick. The coal bunkers are above and below the protective deck, and have a capacity of 200 tons. The coal supply can be increased to 400 tons when necessary.

The armament differs somewhat from that of the *Haarfagre*, and consists of two 8-inch, six 4.7-inch, six 12-pounder and six 1½-pounder rapid fire guns and two 18-inch torpedo tubes.

These vessels have two double-ended and one single-ended cylindrical boilers, the working pressure being 155 pounds per square inch. The expected speed, with assisted draft, is 16 knots. The machinery was built by Messrs. Hawthorne, Leslie & Co., Newcastle-on-Tyne.

In appearance these vessels are similar to the ordinary cruiser or battleship, their freeboard being rather high. All guns are mounted above the upper deck.

Delfin, Hai and **Hval.**—These first class torpedo boats, mentioned on page 200 of Volume VIII, have been delivered to the

Norwegian government by the builder, Mr. F. Schichau, Elbing, Germany.

The length is 127.9 feet; beam, 15.74 feet; displacement, variously given as 80 and 95 tons; designed speed, 23 knots. The armament consists of two 1.46-inch rapid fire guns and two 18-inch deck torpedo tubes. Thornycroft boilers are installed.

Running at 362 revolutions per minute, the engine of one of these boats has developed about 1,300 I.H.P., and given the boat a speed of 24.6 knots.

RUSSIA.

Pereswich and **Oslabya.**—These armored cruisers, building at the Baltic Works, St. Petersburg, and described on page 200 of Volume VIII, are to have three screws. The following particulars are from the "Marine Rundschau," Berlin.

Each of the three triple expansion engines has been designed to develop 4,834 I.H.P. when making 115 revolutions per minute, with a steam pressure of 160 pounds per square inch. The machinery is to be tried under full power during a six hours' run at sea. The total weight of the machinery, not including boilers, must not exceed 860 tons.

The cylinders are to be 38.19, 66 and 80 inches in diameter, The total condensing surface will be about 21,750 square feet. The main steam pipes are to be wire-wound. All elbows in pipes over 7.87 inches in diameter are to be of cast bronze. There will be an evaporator in each engine room, with a capacity of 30 tons of water per day.

There will be 30 Belleville boilers, type 1894, arranged in 6 groups, and having a total grate surface of about 1,393, and a heating surface of about 43,450 square feet. The weight of the boilers is to be 876 tons, with a permissible variation of 45 tons. There will be twelve feed pumps, eight air compressing pumps and twelve blowers. Two of these blowers will be in each fire room, each capable of discharging about 20,000 cubic feet of air per minute when running at 500 revolutions.

Rossija.—This triple screw cruiser, the grounding of which was described on page 186 of the February number, was floated after fifty-one days of hard and almost incessant work, and

steamed safely into Cronstadt. The following abstract of the means adopted to get the cruiser off is translated from the "Marine Rundschau," Berlin, which got its account from the "Morskoi Sbornik."

On November 7, 1896, the day on which the cruiser ran aground, her displacement was about 10,800 tons, the draught on leaving St. Petersburg being 23.95 feet forward and 23.62 feet aft. The strong westerly gale had increased the normal depth of the water by about 3.5 feet. About twelve hours after grounding, the gale moderated and the water fell, soundings then made giving a depth of water around the cruiser of from 18 to 20 feet.

Attempts made during November 7 and 8 to haul the cruiser off showed that this method would be useless. As the vessel was not yet completed, only a few heavy objects, weighing together about 1,045 tons, could be taken out and lightening the cruiser 1.6 feet.

It was then decided to dredge a channel along one side of the cruiser and extend this to deep water. The details of this plan were changed from time to time, the final result, after sixteen days' work, being a channel about 1,330 feet long, and varying in depth from 27 to 28 feet; its width along the port side of the cruiser was about 69 feet; at the bow, where the channel turned at an angle, it increased to 210 feet, and was 174 feet the rest of the way to deep water. The 1,045 tons of weight were removed, and work on the dredging commenced late on November 9. Several suction and lifting dredgers were employed from time to time, but their work was very much interfered with by the formation and movement of the ice and the unfavorable weather.

On November 23, examination by divers showed that the cruiser was then resting on the ground on the whole of her starboard side and on part of the port side amidship. The ends of the cruiser were in water from 25 to 28 feet deep, and the sides of the supporting ground were nearly vertical. It was, therefore, thought possible that the cruiser might be hauled into the deep water, and the *Admiral Senjavin* and *Admiral Oushakoff* were made fast to the bow and stern, respectively. Although the cruiser was heeled over 4°, she did not move towards the hole

that had been dredged. The heeling caused, however, much of the ground supporting the cruiser to slide into the hole, from which the dredgers easily removed it.

On November 25 and 27, the two warships hauled on the cruiser in the direction of the bow, assisted by the cruiser's engines, but without success, and the two ships were then sent into port. The heavily compressed sand still remaining under the keel and starboard side of the cruiser, resisted all attacks of the suction dredgers, and it was decided to change one of the suction dredgers to a forcing dredger and wash away the ground. On December 2, the first attempt was made, but was stopped by the movement of the ice, after one-half hour's work. The results, on examination, were shown to be encouraging. It was now decided to wait until the broken ice alongside the cruiser had frozen solid, as it was not advisable to run any risks of accident to the one available dredger which could do the washing. The channel to deep water ahead of the cruiser was kept open by an ice breaker.

On December 6, a board of officers decided that the work should be continued in spite of the weather conditions, and that, in case it became necessary to have the cruiser winter in the ice, nothing more than wooden shields would be required to protect her hull.

On December 8, the washing process was resumed. On December 16, the vessel had heeled to port $2\frac{1}{2}°$. On December 21, while some repairs were being made to the forcing dredger, an examination showed that the ground under the midship part of the cruiser had several large holes in it, from 20 to 26 feet in diameter, and extending under the starboard side. The heel of the cruiser was $2\frac{3}{4}°$.

As it now seemed likely that the cruiser could soon be hauled off, preparations were made for this purpose. Three wooden capstans were erected on the ice, which was from 12 to 13 inches thick. One block of a heavy tackle was secured to the foot of each capstan and the other block to the cruiser. On the evening of December 23, the heel was $6\frac{3}{4}°$. On December 26, the

greater part of the keel was clear and the ground remaining under the cruiser was full of holes and much loosened. The channel was now kept clear of ice, and high water awaited.

On the evening of December 26, the water began to rise, so that by midnight it was nearly 6 feet above the normal, and the ship righted herself. The windlasses were immediately manned and heaving-in was begun at 1·30 a. m. The ship began to move, slowly she came towards the channel which had been dug out, and finally floated. The long and arduous work of fifty-one days was successfully completed, and the cruiser steamed into port next morning. An examination of the vessel showed that no damage of any kind had been done, rudder and propellers being in perfect condition. The cost of releasing the cruiser amounted to about $19,500, a very small sum under the circumstances.

Standard.—A description of this yacht was given on page 434 of Volume VII, and the result of the trial on page 189 of the February number. The following additional data of the machinery are from the "Marine Rundschau," Berlin.

The twin screw triple expansion engines were designed to develop 10,600 I.H.P. when making 92 revolutions per minute, with a steam pressure, at engines, of 165 pounds per square inch. The cylinders are 41, 65 and 105 inches in diameter, the stroke being 54 inches. The weight of machinery, with water in boilers, is 1,460 tons.

The main condensers are of cast iron and form part of the support for the back of the engines. The total cooling surface is 18,000 square feet. There is also an auxiliary condenser of brass, with a cooling surface of 1,500 square feet. The circulating pumps have a discharge capacity of about 1,000 tons of water per hour at 300 revolutions per minute. The feed and reserve tanks have a capacity of 150 tons. Each engine room is supplied with a feed water heater and a purifier. There are two evaporators, which can supply 60 tons, and one distilling apparatus, which can supply 3,000 gallons of fresh water per day.

The twenty Belleville boilers have a total heating surface of 35,000 square feet, the working pressure being 245 (?) pounds

per square inch. All steam pipes over 7 inches in diameter are wire-wound.

There are three dynamos for the lighting service.

SPAIN.

Cardenal Jimenez de Cisneros.—This twin screw armored cruiser, built at Ferrol, Spain, was successfully launched on March 19. The following data are from "Le Yacht," Paris.

The length over all is 364.2 feet; beam, extreme, 65.14 feet; draught, mean, 21.49 feet; displacement, 6,889 tons.

There is a partial armor belt, extending for about 312 feet of the length of the vessel, and varying from 8.4 to 12 inches in thickness. The protective deck is 1.97 inches thick on the horizontal part, and 2.76 inches on the inclined sides.

The armament will consist of two 11.02-inch guns in barbette turrets, which are from 9.84 to 11.81 inches thick; ten 5.51-inch rapid fire guns in the superstructure, with 1.97-inch shields; two 2.95-inch, and sixteen 2.24- and 1.46-inch rapid fire guns; two machine guns; and eight torpedo tubes, two of which are under water.

The triple expansion engines are to develop 9,863 I.H.P. under natural draft, and about 13,808 I.H.P. under forced draft, the speed of the vessel being 18 and 20 knots, respectively. There are six cylindrical boilers, four of which are double, and two, single-ended.

The full coal supply is about 1,180 tons, and the complement will be 500.

Marques de la Victoria.—This torpedo gunboat, built by Messrs. Vila & Co., Ferrol, was launched on March 5. She is a sister vessel of the *Dona Maria de Molina*, described on page 855 of Volume VIII, although the dimensions published are slightly different, and the two heaviest guns are said to be 5.9-inch instead of 4.7-inch caliber.

The length is 232.9 feet; beam, 27.06 feet; draught, 9.09 feet; displacement, 812 tons. The hull is of Siemens-Martin steel made in Spain. There will be two masts.

The armament will consist of two 5.9-inch Hontoria guns, four 1.65-inch Nordenfeldt rapid fire guns, two machine guns and four torpedo tubes.

The twin screw engines were built at Barcelona, and are to develop about 4,537 I.H.P. and give the vessel a speed of 20 knots.

The total cost of the vessel will be about $400,000.

Audaz, Osado, Pluton and **Proserpina.**—These are the four torpedo boat destroyers building by the Clydebank Engineering and Shipbuilding Co., and described on page 192 of the February number. The *Audaz* was successfully launched on February 6, and the *Osado*, on March 16.

MERCHANT STEAMERS.

Pennsylvania.—This large steamer of the Hamburg-American Steam Navigation Company, some particulars of which were given on page 862 of Volume VIII, arrived at New York on her first voyage in February. For the following particulars of the vessel and her machinery, the JOURNAL is indebted to Mr. Thos. H. Barrett, U. S. Local Inspector of Steam Vessels at New York.

The gross tonnage of the vessel is 13,726.48, and the net tonnage, 8,901.56 tons. The hull is of steel, and is divided by eight watertight bulkheads. She is allowed to carry 25 first and 25 second class, and 100 steerage passengers, although the number of staterooms and the accommodations are for a much larger number of passengers. The complement of officers and men is 105. There are fourteen boats, twelve of which are metallic life boats, and eight Chambers' life rafts.

The twin screw engines are of Schlick's quadruple expansion type, with cylinders 23, 33, 48 and 89 inches in diameter, the stroke being 54 inches. The usual number of revolutions per minute is about 75.

There are five cylindrical steel boilers, each 13 feet 8 inches in diameter. Three of these are double-ended, each 20.5 feet long, and two, single-ended, each 10.5 feet long. The shell plates are 1.5-inches thick, all seams being treble-riveted with 1.5-inch rivets. There are twenty-four furnaces in all, the length of each grate being 5.5 feet, and the width, 3 feet 4 inches. The total grate surface is 440, and the heating surface, 15,020 square feet. The total number of boiler tubes is 1864, all 3.25 inches in diameter. The length of the tubes in the double-ended boilers is 8 feet $4\frac{5}{8}$ inches, and that in the single-ended, 7 feet $3\frac{7}{8}$ inches. In the crown of the center combustion chamber in each boiler there is a fusible plug. The allowed working pressure is 210 pounds per square inch. On each of the double-ended boilers there are three

4⅛-inch, and on each single-ended boiler, two 3⅝-inch safety valves. There are seven feed pumps, four worked by the main engines, two Weir's and one auxiliary duplex, and one injector. The main engine feed pumps are 4.5 inches in diameter and have a stroke of 30 inches; the Weir pumps are 8.5 inches in diameter, with a stroke of 26 inches; the auxiliary duplex pump is 6 inches in diameter with a stroke of 10 inches.

"Marine Engineering," New York, gives the length of the vessel as 560 feet; beam, 62 feet; depth, 42 feet; load displacement, 23,400 tons, at a mean draught of 31.5 feet; bunker capacity, 1,450 tons. The engine room is 38 feet long, and its width, about 62 feet, while the distance from the starting platform to the top of the skylight is 74 feet.

The cylinders follow in the order H.P., L.P., 2d I.P., and 1st I.P., the H.P. cylinder being forward. The angle between the H.P. and 1st I.P. cranks is 60 degrees, between the 1st and 2d I.P. cranks, 200 degrees, and between the 2d I.P. and L.P. cranks, 100 degrees, the H.P. and L.P. cranks being at an angle of 160 degrees to each other. The L.P. cylinder has a slide valve, and the others, piston valves. The valve chests of the L.P. and the 2d I.P. cylinders face each other, and those of the H.P. and 1st I.P. are on the ends.

The valves are worked by the ordinary Stephenson link. The frames are of the box pattern. The crank shaft is built up and is 14¼ inches in diameter in the journals, and 14½ inches in the pins. The air and bilge pumps form part of the back frames and are worked by levers off the L.P. crosshead. In the center of the length of the crank shaft, the turning gear is fitted, and by a simple adjustment of the worm wheels it is operated by the reversing engine, when the vessel is in port. The condensers are of compact build and contain 7,280 square feet of cooling surface.

The diameter of the three-bladed propellers is 16, and the pitch, 21 feet, the port propeller being 5 feet forward of the starboard one, and the horizontal distance between their centers, 13 feet. At the forward end of the engines, the shafts are 17.5 feet apart, their total length being 227 feet. All shafting is hollow,

the line shafts being 13¾, and the propeller shafts, 14½ inches in diameter. The single straight smoke pipe is 12 feet in diameter and its top is 98 feet above the grates.

A Weir's feed water heater receives the exhaust from the auxiliary engines, and is designed to raise the temperature of the feed to boiling point. In the recess between the thrust bearings, at a slightly lower level than the engine room floor, the electric light plant is fitted. There are three Allen's direct-driven dynamos, two of which are in regular use when the vessel is at sea, and the third in reserve. They run at 275 revolutions per minute, each generator supplying current to 300 lamps of 16-candle power. There is also a compact Hall refrigerating machine here which maintains a low temperature in the provision rooms aft. See ash ejectors are fitted, besides the regular ash hoists.

There are in all 45 engines on board, containing 85 steam cylinders.

To show the enormous carrying capacity of this steamer, it may be interesting to note that when the *Pennsylvania* left New York on her first eastward passage, on February 18, she carried 18,500 tons measured cargo, or about 12,000 tons dead weight, which would represent the load carried by about 616 ordinary freight cars. In this cargo, there were about 6,847 tons of grain, a very large cargo by itself. The draught on leaving New York was 31.5 feet.

Friedrich der Grosse.—This twin screw freight steamer of the North German Lloyd Steamship Company, was partially described on page 861 of Volume VIII and on page 201 of the February number. For the following additional particulars, the JOURNAL is indebted to the courtesy of Mr. Thomas H. Barrett, U. S. Local Inspector of Steam Vessels at New York.

The gross tonnage of the vessel is 10,536, and the net tonnage, 6,765 tons. The steel hull is divided by twelve watertight bulkheads. The draught is 27 feet. She has accommodations for and is allowed to carry 202 first class, 78 second class and 646 steerage passengers, besides her complement of 174 officers and men. There are fourteen metallic life boats and six Chambers' life rafts.

The twin screw, quadruple expansion engines have cylinders

25¼, 38 3/16, 52½ and 75⅝ inches in diameter, the stroke being 55¼ inches. The engines run usually at 75 revolutions per minute.

There are seven main and one auxiliary boilers, all cylindrical and made of steel. Of the main boilers, five are double-ended, each 19 feet 1½ inches long, and two, single-ended, each 10 feet 8¾ inches long, the diameter of all being 13 feet 8 inches. The double-ended boilers have four furnaces each, and the single-ended, two furnaces each, the grates being 6 feet 6 inches long and 3 feet 11 inches wide. The total grate surface of the main boilers is 611 square feet, and the allowed steam pressure, 213 pounds per square inch. The shell plates are 1⅛ inches thick, longitudinal seams being double butt-strapped and triple-riveted, and the circumferential seams triple-riveted, the rivets being 1½ inches in diameter. The total number of 3-inch tubes in the main boilers is 3,840, and their length, 7 feet 2⅛ inches. The usual fusible plug is fitted in the top of one of the combustion chambers in each boiler. On each double-ended boiler there are four, and on each single-ended boiler, two 3⅝-inch safety valves. There are four feed pumps worked by the main engines, four independent Weir's and two auxiliary duplex feed pumps. The engine pumps are 4⅜ inches in diameter and have a stroke of 23⅝ inches, the Weir pumps, 8 inches in diameter with a stroke of 21 inches, and the duplex pumps, 5½ inches in diameter with a stroke 17¾ inches. There are four 5½-inch bilge pumps, two worked by the main engines, and two, independent, the stroke of the engine pumps being 23⅝, and of the others, 17¾ inches. There is besides a bilge ejector. There are three duplex fire pumps, of the same size as the independent bilge pumps. Four sanitary pumps, which may be used as fire pumps, are worked by the main engines; diameter 4¼, and stroke, 23⅝ inches.

The auxiliary boiler on the main deck is 7 feet 10½ inches long and 7 feet 2½ inches in diameter, and has one furnace, 5 feet 3 inches long and 31½ inches in diameter, the grate surface being 13.77 square feet.

Oceanic.—This steamer, building for the White Star Line by Messrs. Harland & Wolff, Belfast, Ireland, will be the largest steamer so far proposed. The only definite information so far

obtainable is that the length over all will be 704 feet, and between perpendiculars, 685 feet; tonnage about 18,000. There will be two funnels and three masts. She will be fast, but not extremely so, and, in all probability, will have twin screws.

Raglan Castle.—This steamer, built by Messrs. Barclay, Curle & Co., Clydeholm, Whiteinch, for the Castle Mail Packet Co., was launched on January 20 and finished her trials on March 16. She is practically a duplicate of the *Dunolly Castle*, described on page 205 of the February number, except that she is 15 feet longer, thus increasing her gross tonnage to 4,320 tons.

The triple expansion engine was designed for about 2,200 I.H.P. The contract speed was 12.5 knots, but on the trial, under unfavorable weather conditions, a speed of 13.75 knots was attained.

Princess Anne.—This steel steamer, built by the Delaware River Iron Shipbuilding and Engine Works, Chester, Pa., for the Old Dominion S. S. Co., was successfully launched on May 1. Some particulars of the proposed steamer were given on page 200 of the February JOURNAL, but the following data are correct for the finished vessel.

The length is 323 feet; beam, 42 feet; depth, molded, 27 feet; tonnage, 3,300. There are accommodations for 120 first class and 25 steerage passengers.

The single engine is of the triple expansion type, designed to develop about 3,400 I.H.P. under natural draft, and give the vessel a speed of 17 knots. The cylinders are 27, 44.5 and 73 inches in diameter, the stroke being 54 inches. There are four boilers, each 14.25 feet in diameter and 12 feet long, with a total grate surface of 300, and heating surface of 10,000 square feet. There is one smoke pipe, its top being 80 feet above the grates.

The four-bladed propeller is 16 feet in diameter and has a pitch of 23 feet.

————.—This steel sheathed steamer is building by Messrs. F. W. Wheeler & Co., West Bay City, Mich. The length over all will be 281 feet, between perpendiculars, 266 feet; beam, 42 feet; depth, 19 feet; depth of ballast tank at center, 3 feet 6 inches; carrying capacity, 2,500 net tons on a mean draught of 12 feet

10 inches. The vessel will be single-decked, with bulwarks 4 feet high fore and aft. The plates used in the construction will be of the best open-hearth mild steel, with a limit of tensile strength of from 54,000 to 62,000 pounds per square inch. There will be seven cargo hatches. The hull, up to the turn of the bilge, will be sheathed with 5 inches of oak. The engine will be of the vertical triple expansion type, with cylinders 17, 29 and 48 inches in diameter, the stroke being 36 inches. There will be two cylindrical boilers, each 11 feet in diameter and 12 feet long, built for a working pressure of 170 pounds per square inch. Each boiler will have two plain furnaces, 42 inches in diameter. There will also be an auxiliary boiler. Steam steering gear, electric lighting plant, Dean Bros.' pumps, and steam windlasses and capstans will be installed.

Duchess of Devonshire.—This fast twin screw steamer, built by the Naval Construction & Armaments Co., Barrow, for the service between Belfast, Barrow and the Isle of Man, completed her trials under very unfavorable weather conditions. The following data are from "Engineering," London.

The length is 300 feet; beam, 35 feet; depth, molded, 15 feet; draught (on trial), 10 feet 3.5 inches; corresponding displacement, 1,720 tons; tonnage, gross, 1,280, net, about 500 tons. She carries 120 first class, and may carry 1,200 steerage passengers. Deep bilge keels are fitted.

The triple expansion engines were designed to develop 5,000 I.H.P. when making 175 revolutions per minute. The cylinders are 21, 32 and 50 inches in diameter, the stroke being 30 inches. The cylinders are separate castings, bolted together, supported amidships on cast iron columns, and at the sides of the ship by wrought steel columns, which are secured to strong cast iron bedplates. The shells of the condensers are formed of wrought steel plates, and the total cooling surface is 5,400 square feet.

There are two double-ended boilers, each 14.5 feet in diameter and 20.25 feet long, the working pressure being 200 pounds per square inch. The total grate surface is about 257, and the heating surface, about 8,900 square feet. Forced draft may be used.

The propeller hubs are of cast steel and the blades of bronze, the propellers being 9.75 feet in diameter.

The trials consisted of progressive runs over the Skelmorlie mile at 12.5, 15, 16.75 and 18.25 knots; these runs were for the information of the contractor only. Next, forced draft runs were made, the first run giving a speed of 19.56, and the mean of two runs, over 19 knots. A six hours' forced draft trial was then made, a speed of 18.8 knots, with one inch air pressure, being attained. The vessel proved herself a good sea boat during the very bad weather on the trials.

————.—This new twin screw freight steamer is building by the Forges et Chantiers de la Méditerranée at La Seyne for the Messageries Maritimes Co. The following particulars are from "Le Yacht," Paris.

The length between perpendiculars will be 393.7 feet; beam, 45.28 feet; depth, 33.3 feet; mean draught, 22.01 feet; displacement, light, without coal, 3,503 tons; load displacement, 7,337 tons; gross tonnage, about 4,429 tons.

The hull will be the first one built in France with joggled plates, used so extensively by Messrs. Wm. Doxford & Sons, Sunderland, England. There will be six watertight bulkheads and four principal compartments in the double bottom. There will be two masts and one smoke pipe.

The twin screw engines will be of the triple expansion type, developing a total of 3,255 I.H.P. at their lowest rate of revolution. Slide valves and Marshall gear will be used. There will be three double-ended boilers with a working pressure of 128 pounds per square inch. Two of the boilers will have six and the other four furnaces. The propeller blades will be of bronze and the boss of steel.

YACHTS.

Marietta.—This yacht, built by the John N. Robins Co., Erie Basin Dry Docks, Brooklyn, N. Y., and described on page 207 of the February number, was successfully launched on March 27.

Hiawatha.—This yacht, built by the Gas Engine and Power Co., New York, and described on page 208 of the February number, was successfully launched on April 17.

Margarita II.—This yacht, mentioned on page 211 of the February number, was there incorrectly named *Marietta II.*

Alcedo.—This yacht, built by the Harlan and Hollingsworth Co., for Mr. G. W. Childs Drexel, and partially described on page 211 of the February number, was successfully launched on April 10. For the following description, the JOURNAL is indebted to Mr. T. Jackson Shaw, Superintending Engineer of the Company.

The yacht is of steel and built to conform to the requirements of the U. S. Standard Rules. Schooner rig with pole masts.

The length over main rail is 174 feet 8 inches, on deck, 170 feet 8 inches, and on water line, 150 feet 8 inches; beam molded, 24 feet; depth, 14.75; and draught extreme, 11 feet.

There is a steel house on the main deck containing a dining saloon, pantry, galley, engine and boiler rooms, chief engineer's room and a smoking room. The deck of this house forward will be carried out to the rail, forming the bridge. On top of this house, and forward of the smoke pipe, there is a chart room and a room for the captain. The owner's and guests' rooms are below the main deck and are fitted up in hard woods of different kinds and handsomely decorated. The dining saloon will also be fitted in hard wood in an elaborate style. The officers and crew are located forward below deck.

The motive power will consist of a three-cylinder, inverted triple expansion engine, with independent condenser, air and feed

pumps. The style of engine is the same as now being used in torpedo boat practice, having forged steel column supports with diagonal braces. The cylinders are 14.5, 24 and 37 inches in diameter, the stroke being 22 inches. The propeller is a true screw of manganese bronze, having a diameter of 7.5, and a pitch of 10 feet.

The three boilers are of the Almy water tube type, having a total grate surface of 87 square feet, and designed for a working pressure of 225 pounds per square inch. The engine, however, will work under a pressure of about 200 pounds per square inch.

The yacht will be fitted throughout with electric lights and have a powerful search light, electric bells, baths, toilet rooms, &c., in first class yacht style and finish, and is expected to attain a cruising speed of 13½ knots, with a maximum speed under forced draft of 15 knots.

Ellide.—This yacht, built by Mr. Samuel Ayres, at Nyack, for Mr. E. Burgess Warren of Philadelphia, from designs by Mr. C. D. Mosher of New York, was launched on April 17. The following particulars are from "Seaboard," New York.

The length is 80 feet; beam, 8 feet 4 inches; draught, 3.5 feet. The yacht is of composite construction, the planking being of double-skin mahogany, all fastening bolts being of bronze. She is subdivided by five steel bulkheads into six water tight compartments, and is supplied with a number of copper air tanks, so that, if filled with water, she would still float. The Mosher patent quadruple expansion engine is designed to develop between 600 and 800 I.H.P. The cylinders are 9, 13, 18 and 24 inches in diameter, the stroke being 10 inches. All the reciprocating parts of the engine are hollow, and all steel used in its construction is oil-tempered. There is a surface condenser. Six specially designed engines drive the feed pumps, air pumps, and circulating pump. The total weight of the engine is about 2 tons. The boiler is of the Mosher water tube type, its outside dimensions being 6 by 8 by 4.5 feet, and its weight, about 3 tons. It consists of two steam drums placed over two water drums and connected together by 1,228 solid-drawn steel tubes, so arranged that the gases pass twice the length of the boiler among the tubes before

entering the smoke pipe. The boiler is subdivided into two independent sections, so that in case a tube should give out in either section, the remaining section can still be used.

Ægusa.—This large steel yacht, designed and built by Messrs. Scott & Co., Greenock, Scotland, for Mr. Ignazio Florio of Palermo, Italy, made her official trial on February 27.

Length over all, 264 feet, between perpendiculars, 251.25 feet; beam, 31.5 feet; depth, 20.5 feet; tonnage, Thames yacht measurement, 1,230 tons. There are nine water tight bulkheads. The owner's quarters are forward under the main deck. Aft of the machinery space, there is a large saloon, a dining room and several staterooms. In the deck house there is a smoking room, a reception room, a stateroom for the owner, the galley and the chart house. The crew is quartered forward, and the officers and servants, aft.

The four-cylinder triple expansion engine has cylinders 24, 47.5 and (two) 53.25 inches in diameter, the stroke being 47 inches. The designed I.H.P. was 2,500, with a working pressure of 170 pounds per square inch. There are two cylindrical boilers, each 17 feet 8.5 inches in diameter and 13 feet 11.25 inches long. Each boiler has four Fox's corrugated furnaces, 45.25 inches in mean diameter.

The electric plant consists of two dynamos. Among the boats carried, there are a steam launch and a Daimler motor launch. A Kilbourn refrigerating machinery is installed.

On her trial, a speed of 16.25 knots over the measured mile was attained.

Rosabelle.—This composite yacht, built by Messrs. Ramage & Ferguson, Leith, for Mr. T. Pim of Marten's Grove, Kent, was launched on February 18. The following particulars are from "The Practical Engineer," London.

The length over all is 167 feet, and between perpendiculars, 141.5 feet; beam, 26 feet; depth, 14 feet; tonnage, about 440, yacht measurement. She is rigged as a topsail schooner, and has an outside hanging keel made of 50 tons of lead. The owner's staterooms are forward of the machinery space, with a deck saloon above. Abaft the machinery space is a large dining

room, five other state rooms and a drawing room at the extreme after end. The owner's state rooms, the dining and the drawing rooms will be covered with silk broché tapestry; the other rooms will be finished in hard wood.

The engine is of the inverted, triple expansion type with cylinders 13, 21 and 33 inches in diameter, the stroke being 22 inches. There are two Yarrow boilers, placed side by side, with a total heating surface of over 2,000 square feet. They are to work under natural draft only, and with a pressure of 160 pounds per square inch.

Leon Pauilhac.—This steel yacht, built by Messrs. Ramage & Ferguson, Leith, for Mr. G. Pauilhac of Toulouse, was launched on February 20, and completed her trial in April.

The length between perpendiculars is 144 feet; beam, 21 feet; depth, 12 feet; tonnage, 307. State rooms are arranged forward and abaft of the machinery space, the dining room being forward.

The triple expansion engine has cylinders 13, 21 and 34 inches in diameter, the stroke being 24 inches. The boiler is built for a working pressure of 170 pounds per square inch.

On trial, a speed of 13.5 knots on the measured mile was attained, exceeding the contract requirement.

Andria.—This steel yacht, built by the Ailsa Shipbuilding Co., Troon, for Mr. John E. Brooks of the United States, was launched on February 18. The designs are by Mr. G. L. Watson.

The length on load water line is 161 feet; beam, 23.25 feet; depth, 14.25 feet; tonnage, Thames yacht measurement, about 300 tons. The triple expansion engine, built by Messrs. D. Rowan & Sons, has cylinders 16, 26 and 41 inches in diameter, the stroke being 27 inches.

Latharna.—This steel yacht, the hull of which is practically a duplicate of the *Andria*, was launched on March 31. She was built for Mr. H. H. Smiley by the same builders. The only difference is in the beam, which is 23.5 feet.

The triple expansion engine was built by Messrs. Muir & Houston and has cylinders 14, 22 and 36 inches in diameter, the stroke being 24 inches.

BOOK REVIEW.

CONSTRUCTION PRATIQUE DES NAVIRES DE GUERRE. Par A. Croneau, Ingénieur des Constructions Navales; Professor à L'Ecole du Genie Maritime. Published by Messrs. Gauthier-Villars et Fils, 55 Quai des Grands-Augustins, Paris.

The first volume of this work was reviewed on page 844 of Volume VI of the JOURNAL, and the second volume, which completes the work, has just been received. It is a book of 574 pages of text, containing 359 cuts, and gives the latest European and American practice in warship construction, from the battleship to the torpedo boat. It deals with subdivision, the disposition and fitting of armor, openings in hull above and below water, hull fittings generally, ventilation, water service, rudders, corrosion and preservation, and weight and resistance of hulls. The same thoroughness of detail pervades the second volume as was noticed in the first. The entire work consists of two volumes of text containing 664 cuts, and a book of double quarto-page plates, and is probably the most comprehensive publication of its kind in existence, and well worthy of study by those interested in the subject. It is thoroughly up to date, the illustrations being those of actual ships and their construction and fittings.

LES NAVIRES DE GUERRE, ESSAI SUR LEUR VALEUR MILITAIRE.—R. Berard, Lieutenant de Vaisseau. Published by Messrs. Berger-Levrault & Cie., Paris, 1897.

To obtain a reliable and accurate method by which two ships of the same or of different types can be compared, so that the most efficient type may be chosen, and the money allowed by the appropriations expended in the most advantageous manner, Mr. Berard studies separately each of the elements which constitute the power and its efficiency under different circumstances, and expresses their value in functions of the fundamental units

of force, space and time: *i. e.*, he measures them, weighs them in the full meaning of the words, in order to present the military value of the ship in so many foot-tons.

As the power of a warship depends upon the destructive effect she can produce upon an adversary, moving or stationary, by means of her three offensive weapons, viz: Aërial projectiles (from the 16-inch shell to the small rifle bullet), the ram, and under water projectiles (torpedoes and submarine guns), each of these factors is carefully investigated and formulas deduced for their values under different conditions. The important factor *time* in all its bearings is also investigated and evaluated, and a final formula found for the useful "work" or power that a ship can develop and direct, in a given time, against an adversary. Although not a perfect solution, the author hopes that by his proposed method and more extended study, the best kind of warship for a given purpose can at once be determined.

SCHÄFFER & BUDENBERG,

MANUFACTURERS OF THE

IMPROVED THOMPSON INDICATOR.

Adapted for all speeds, unsurpassed for Reliability and Excellence of Workmanship. Sold at a Moderate Price.

TACHOMETERS.

Pressure Gauges for all Purposes, Engine Counters and Registers, Marine Clocks, Thermometers.

The Peerless and Manhattan Automatic Injectors. Reducing and Regulating Valves, &c.

WRITE FOR CATALOGUE.

Works, BROOKLYN, N.Y.

SALESROOMS:

No. 22 W. Lake Street, Chicago.
No. 66 John Street, New York.

ELEPHANT BRAND
PHOSPHOR-BRONZE.

REG. TRADE MARKS

THE PHOSPHOR BRONZE SMELTING CO. LIMITED,
2200 WASHINGTON AVE., PHILADELPHIA.
"ELEPHANT BRAND PHOSPHOR-BRONZE"
INGOTS, CASTINGS, WIRE, RODS, SHEETS, ETC.
— DELTA METAL —
CASTINGS, STAMPINGS AND FORGINGS.
ORIGINAL AND SOLE MAKERS IN THE U.S.

DELTA METAL.
PROPELLER CASTINGS.

WORTHINGTON PUMPS

FOR

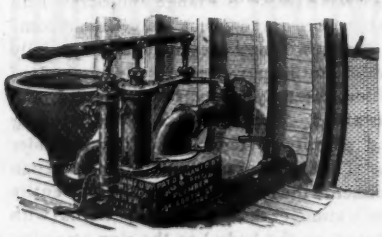
MARINE SERVICE

Horizontal Boiler-Feed Pumps
Vertical Boiler-Feed Pumps
Tank Pumps — Low Service Pumps
Bilge Pumps
Sanitary Pumps
Circulating Pumps — Ballast Pumps
Air and Circulating Pumps
Wrecking Pumps — Independent Condensers

HENRY R. WORTHINGTON
86 and 88 Liberty Street — NEW YORK

PUMP WATER CLOSET

For Yachts, Pilot Boats, Naval Ships, etc.



In Use on U. S. Ships
NEW YORK,
MINNEAPOLIS,
INDIANA,
SAN FRANCISCO,
AND DOLPHIN,
OTHERS,
AND
All Kinds of Yachts and Launches,
From 25 Feet Up to 285 Feet.

FOR ABOVE OR BELOW WATER LINE.

WILLIAM BISHOP'S SONS,

Plumbing, Ventilating, Steam Fitting

AND MARINE SPECIALTIES,

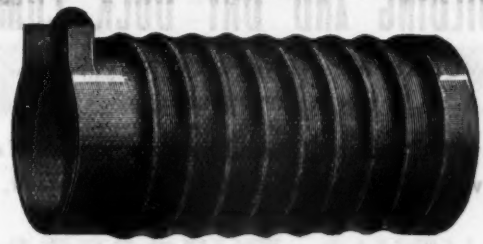
205 South St., New York. — 724 Third Ave., Brooklyn, N. Y.

TELEPHONE CONNECTION.

MORISON SUSPENSION FURNACES,

—FOR—

LAND AND MARINE BOILERS.



UNIFORM THICKNESS — EASILY CLEANED — UNEXCELLED FOR STRENGTH.

Also, FOX CORRUGATED FURNACES.

THE CONTINENTAL IRON WORKS,

P. O. Station G. BROOKLYN, N. Y.

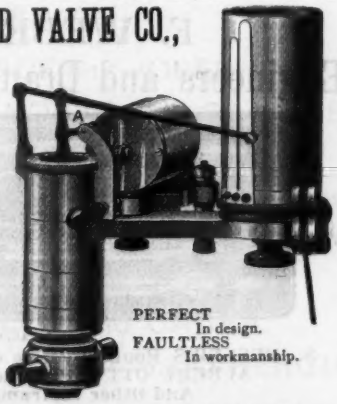
CROSBY STEAM GAGE AND VALVE CO.,

Sole Manufacturers of the

Crosby Steam Engine INDICATOR.

Approved and adopted by the U. S. Government. It is the standard in nearly all the great Electric Light and Power Stations of the United States. It is also the standard in the principal Navies, Government Ship-Yards, and the most eminent Technical Schools of the world.

When required, it is furnished with Sargent's Electrical Attachment, by which any number of diagrams from Compound Engines can be taken simultaneously.



PERFECT
In design.
FAULTLESS
In workmanship.

ALSO SOLE MANUFACTURERS OF

Crosby Improved Steam Gages, Pop Safety Valves, Water Relief Valves, Patent Gage Testers, Safe Water Gages, Revolution Counters, ORIGINAL Single Bell Chime Whistles and other Standard Specialties used on Boilers, Engines, Pumps, etc.

Main Office and Works: Boston, Mass., U. S. A.

Branches: New York, Chicago, and London, Eng.

NEWPORT NEWS

SHIPBUILDING AND DRY DOCK COMPANY.

WORKS AT NEWPORT NEWS, VA.

(ON HAMPTON ROADS.)

Equipped with a Simpson's Basin Dry Dock, capable of docking a vessel 600 feet long, drawing 25 feet of water, at any stage of the tide.

REPAIRS MADE PROMPTLY AND AT REASONABLE RATES.

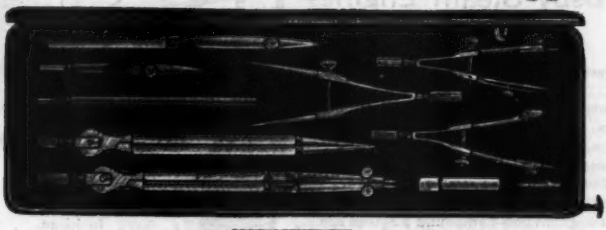
· SHIP AND ENGINE BUILDERS ·

For Estimates and further particulars, address

C. B. ORCUTT, Pres't, No. 1 Broadway, New York.

F. WEBER & CO.,

Engineers' and Draughtsmen's Supplies.



SOLE AGENTS FOR

S. RIEFLER'S Round System of Drawing Instruments, and ALBERT OTT'S Planimeters and Pantographs, And Other Instruments of Precision.

Drawing and Blue Print Papers, Tracing Cloth, Bramwell's Roller T-Square Guide.

DRAWING BOARDS AND TABLES IN LARGE VARIETY.

1125 Chestnut Street, PHILADELPHIA, PA.

BRANCH HOUSES: { **709 Locust Street, St. Louis, Mo.** / **5 North Charles Street, Baltimore, Md.**

SEND FOR CATALOGUE VOL. 110.

LEWIS NIXON,

SHIPBUILDER.

OFFICE AND WORKS,

THE CRESCENT SHIPYARD,

ELIZABETHPORT, N. J.

BUILDER OF

Steam Yachts FREELANCE and JOSEPHINE.
STANDARD OIL BOATS Nos. 77 and 78.
Pennsylvania Ferryboat CAMDEN.
Ten Lake and Canal Barges.
Lake Steamers BETA, GAMMA and DELTA.
U. S. GUNBOAT No. 10.
Sternwheelers RODOLFO and CAURA.
Sidewheeler MARIA HANABERGH.

SPECIAL FACILITIES FOR

REPAIR WORK OF ALL KINDS.

ATTENTION, PLEASE!

Builders of Marine Boilers will be interested to know we are now producing a superior quality of

SEAMLESS CHARCOAL STEEL BOILER TUBES.

These Tubes are made to stand the severest tests. True to diameter and gauge, drawn cold WITHOUT WELD OR SEAM, they commend themselves to manufacturers of boilers requiring the

HIGHEST EFFICIENCY, DURABILITY AND STRENGTH.

Correspondence solicited.

THE SHELBY STEEL TUBE CO.

MAIN OFFICE AND WORKS:
SHELBY, OHIO, U. S. A.

EASTERN OFFICE:
144 Chambers Street, NEW YORK CITY.

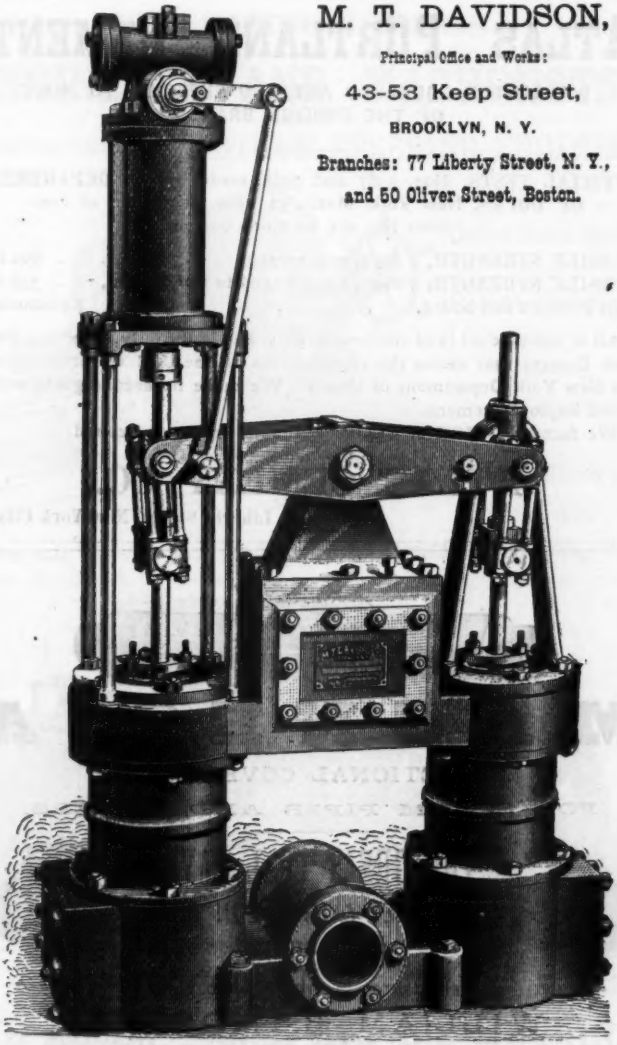
M. T. DAVIDSON,

Principal Office and Works:

43-53 Keep Street,

BROOKLYN, N. Y.

Branches: 77 Liberty Street, N. Y.,
and 50 Oliver Street, Boston.



MARINE AIR PUMP,

AS FURNISHED

U. S. STEAMER "BANCROFT."

ATLAS PORTLAND CEMENT.

WARRANTED EQUAL TO ANY AND SUPERIOR TO MOST OF THE FOREIGN BRANDS.

OFFICIAL TESTS, Nos. 3567 and 3568, made by the **DEPARTMENT OF DOCKS,** New York, March 31, 1894, being part of contract No. 464 for 8,000 barrels.

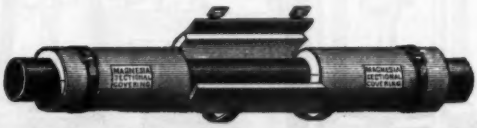
TENSILE STRENGTH, 7 days, neat cement, ... 622 lbs.
TENSILE STRENGTH, 7 days, 2 parts sand to 1 of cement, . 332 lbs.
Pats steamed and boiled, ... Satisfactory.

All of our product is of the first quality, and is the only American Portland Cement that meets the requirements of the U. S. Government and the New York Department of Docks. We make no second grade or so-called improved cement.

We furnish QUICK or SLOW Setting Cement, as desired.

ATLAS CEMENT CO.,

143 Liberty Street, New York City.



MAGNESIA

SECTIONAL COVERING

FOR STEAM PIPES AND BOILERS.

"THE GREAT COAL SAVER."

KEASBEY & MATTISON CO.,

AMBLER, PA.

REPRESENTED IN WASHINGTON, D. C., BY

WILLIAM B. MORGAN,

No. 19, Builders' Exchange.

☞ WRITE FOR ADDRESS OF SELLING AGENT NEAREST TO YOU. ☜

WILLIAMSON BROS.,

ENGINEERS AND MACHINISTS.

MANUFACTURERS OF **STEAM STEERING ENGINES.**

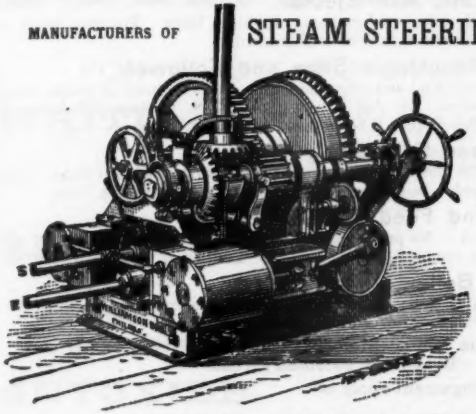
USED IN THE

United States Navy.

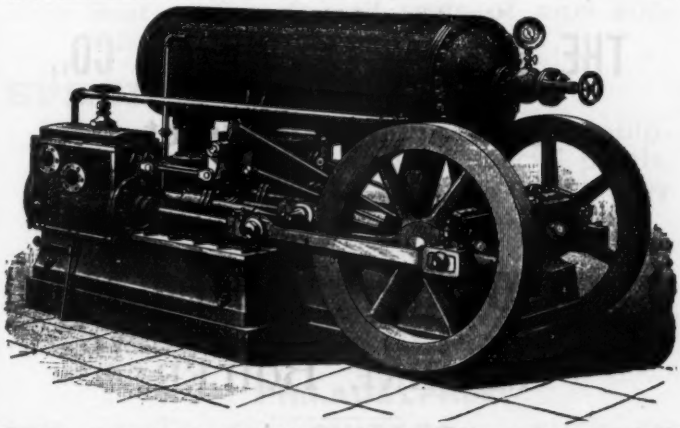
Patent Hoisting and Winding Engines.

Steam Cranes,

&c., &c.



Office and Works: Richmond and York Streets, Philadelphia, Pa.



THE ALLEN DENSE AIR ICE MACHINE.

Contains Air Only. Used on the New Cruisers and Demanded by the Specifications of the Navy Department.

H. B. ROELKER,

No. 41 MAIDEN LANE, NEW YORK.

DESIGNER AND MANUFACTURER OF SCREW PROPELLERS. CONSULTING AND CONSTRUCTING ENGINEER.

SEE HYDRO PNEUMATIC ASH EJECTOR. PISTON WITH ADJUSTABLE SHOE. AIR AND GREASE EXTRACTOR. EVAPORATOR AND FEED WATER HEATER. WATER TUBE BOILER.

1. **Hydro-Pneumatic Ash Ejector.**
   Avoids dust and noise. Saves coal. Lightens labor. Prevents injury to paint. More than 300 now fitted and ordered.
2. **Piston with Adjustable Shoe and Follower.**
   Applicable to horizontal and other engines. Takes the wear at proper point and guides without tail rod. In use on U. S. Cruisers "Baltimore," "Newark," "Philadelphia," Steamer "Connecticut," &c.
3. **Air and Grease Extractor.**
   Prevents grease in boiler. Does not reduce temperature of feed water. Can be quickly cleaned. In use on S. S. "El Rio," "Morgan City," &c.
4. **Evaporator and Feed Water Heater.**
   Rapid in its work. No pump required. Coils readily shed all deposit. Of light weight. In use on S. S. "El Norte," "Louisiana," Yacht "Almy," &c.
5. **Water Tube Boiler.**
   Straight sub-vertical tubes of small diameter. Safe, light and durable. No cast metal. Combined induced and forced draft.

All patented in the United States. The Ash Ejector, Boiler and Evaporator patented abroad.

In order to avoid infringement apply to

HORACE SEE,

No. One Broadway, - - NEW YORK.

THE BABCOCK & WILCOX CO.,

29 Cortlandt Street,

NEW YORK.

Forged Steel Water-Tube

MARINE BOILER.

Light, durable, easily cleaned and inspected. Only straight tubes are used.

The largest cargoes on the Lakes are carried by steamers using BABCOCK & WILCOX Boilers.

The fastest of the six composite Gunboats recently constructed for the U. S. N. ("Marietta" and "Annapolis") are equipped with these boilers.

Boilers also ordered for U. S. S. "Chicago" and "Atlanta."

PAINT

OF ALL KINDS
FOR EVERY POSSIBLE PURPOSE.

WHITE LEAD

RED LEAD

PAINTERS' COLORS

ZINC WHITES

French and American. Guaranteed strictly pure and of best make and preparation.

ANTOXIDE.

A perfect preservative for structural iron. May be applied to hot or cold surfaces. Best coating for all sorts of pipe surfaces for all exposures.

SPECIAL PAINTS

Large factories and complete equipment permit the supply of special paints, made to any given formula, and in any quantity, at shortest notice, with rigid

ADHERENCE TO SPECIFICATIONS GUARANTEED.

HARRISON BROTHERS & CO.

PHILADELPHIA
CHICAGO
NEW YORK

Correspondence solicited on any paint topic.

THE

STRATTON SEPARATOR.

This is the only apparatus that automatically separates water from steam, and secures the maximum of Economy, Efficiency and Safety.

Specified by the Navy Department, and used on all the New Cruisers of the

UNITED STATES NAVY,

and by the Leading

Steamship Lines, Railroads, Water Works and Electric Light Companies.

THE GOUBERT MFG. CO.,

(Sole Manufacturers),

14 and 16 Church St., cor. Cortlandt, New York.

WESTINGHOUSE ELECTRIC AND MFG. COMPANY,

PITTSBURG, PA.

The Largest and Most Completely Equipped Electrical Manufacturing Establishment in the World.

Power, Incandescent and Arc Lighting, from the Same Circuits.

We have purchased and are the SOLE OWNERS of the patents issued to Nikola Tesla for the POLYPHASE ALTERNATING SYSTEM, now recognized to be the most successful system for lighting and power purposes.

We furnish complete lines of apparatus for the perfect equipment of ISOLATED PLANTS for Hotels, Office Buildings, Flats, Factories, as well as STEAMERS.

STANDARD SYSTEMS for distribution of lights and power in LARGE MANUFACTURING ESTABLISHMENTS, MILLS and MINES.

WESTINGHOUSE ELECTRIC RAILWAY SYSTEM,

The Most Durable, Economical and Efficient on the Market.

NEW YORK, 120 Broadway.
BOSTON, Exchange Building.
CHICAGO, New York Life Building.
TACOMA, WASH., 102 S. Tenth Street.
PITTSBURG, Westinghouse Building.
BUFFALO, Erie County Bank Building.
PHILADELPHIA, Girard Building.
ST. LOUIS, American Central Building.
CHARLOTTE, N. C., 36-38 College Street.
SYRACUSE, N. Y., Bastable Building.
SAN FRANCISCO, Mills Building.
WASHINGTON, D. C., 1333 F Street, N. W.
For CANADA, address AHEARN & SOPER, Ottawa, Canada.

W. & A. FLETCHER CO.,

North River Iron Works,

MARINE ENGINES, BOILERS, ETC.

Hudson, 12th and 14th Streets,

HOBOKEN, N. J.

TAKE FERRY FROM FOOT OF WEST 14TH ST., N. Y.

ASBESTOS FIRE-FELT

FOR COVERING

MARINE • BOILERS.

SECTIONAL,
DURABLE,
REMOVABLE,
FIRE-PROOF,
WATER-PROOF.

Approved by U. S. NAVY DEPARTMENT, and in use on the NEW CRUISERS of the U. S. NAVY.

H. W. JOHNS Asbestos Roofing, Sheathing, Building Felt, Asbestos Steam Packings, Boiler Coverings, Roof Paints, Fire-Proof Paints, etc.

VULCABESTON Moulded Piston-Rod Packing Rings, Gaskets, Sheet Packing, etc.

H. W. JOHNS MFG. CO., 100 William St., New York.

CHICAGO. PHILADELPHIA. BOSTON. LONDON.

ORFORD COPPER CO.

37 Wall Street, New York.

ROBERT M. THOMPSON, Prest.

COPPER INGOTS, WIRE BARS AND CAKES

FERRO NICKEL

AND

FERRO NICKEL OXIDE

FOR USE IN PREPARING

NICKEL STEEL FOR ARMOR PLATES.

R. BERESFORD,

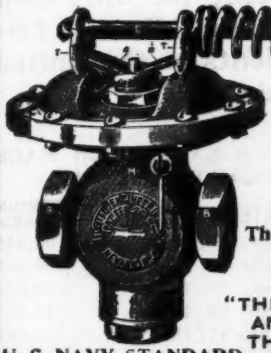
PRINTER AND BINDER,

617 E STREET, N. W.,

CITY OF WASHINGTON.

Journal of American Society of Naval Engineers Bound in Library Style for $1.50 per Volume.

The FOSTER



"New Class W"

Pressure Regulator.

(Reducing Valve)

The Experimental Board of the Bureau of Steam Engineering of the U. S. Navy, after recent tests, report that it is:

"THE BEST PRESSURE REGULATOR AND REDUCING VALVE WITHIN THEIR KNOWLEDGE."

U. S. NAVY STANDARD.

FOSTER ENGINEERING CO.

(FOR SHIPBUILDERS) NEWARK, N. J.

GEIPEL'S PATENT STEAM TRAP.

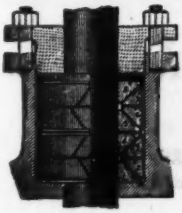
GUARANTEED STEAM TIGHT.

THORPE, PLATT & CO., 97-103 Cedar St., N. Y.

VALVE CAN BE GROUND IN PLACE.

Can be entirely taken apart, cleaned and put together again in 5 minutes.

KATZENSTEIN'S Self-Acting Metal Packing.



For PISTON RODS, VALVE STEMS, etc., of every description, for Steam Engines, Pumps, etc., etc.

Adopted and in use by the principal Iron Works and Steamship Companies, within the last twelve years, in this and foreign countries.

FLEXIBLE TUBULAR METALLIC PACKING, for slip-joints on Steam Pipes, and for Hydraulic Pressure; also METAL GASKETS for all kinds of flanges and joints.

DOUBLE-ACTING BALANCED WATER-TIGHT BULKHEAD DOORS for Steamers. Also Agents for the McColl-Cumming PATENT LIQUID RUDDER BRAKE.

For full particulars and reference, address,

L. KATZENSTEIN & CO.,

General Machinists, Brass Finishers, Engineers' Supplies,

357 West Street, New York.

The POCAHONTAS COAL

Is the Best Steam Fuel Mined, and enjoys the Unique Distinction of being the Only Coal in the World that has been Officially Endorsed by the Governments of the United States and Great Britain.

WAR SHIPS OF THE UNITED STATES NAVY WHICH MADE THEIR TRIAL TRIPS WITH POCAHONTAS COAL.

VESSEL.	BUILDERS.	DATE OF TRIAL.	PREMIUM EARNED.
Baltimore,	Cramp & Sons,	Nov. 15, 1889	$106,442
Philadelphia,	Cramp & Sons,	June 25, 1890	100,000
Newark,	Cramp & Sons,	Dec. 22, 1890	36,857
Bancroft,	Moore & Sons,	Jan. 26, 1893	45,000
Detroit,	Columbia Iron Works,	Apl. 17, 1893	150,000
New York,	Cramp & Sons,	May 22, 1893	200,000
Machias,	Bath Iron Works,	June 10, 1893	45,000
Castine,	Bath Iron Works,	Sept. 15, 1893	50,000
Columbia,	Cramp & Sons,	Nov. 18, 1893	350,000
Marblehead,	Quintard Iron Works,	Dec. 7, 1893	125,000
Montgomery,	Columbia Iron Works,	Jan. 19, 1894	200,000
Minneapolis,	Cramp & Sons,	July 14, 1894	400,000
Maine,	Navy Yard, New York,	Oct. 17, 1894	29,200
Indiana,	Cramp & Sons,	Oct. 18, 1895	50,000
Texas,	Navy Yard, Norfolk, Va.,	Dec. 19, 1895	1,000
Massachusetts,	Cramp & Sons,	Apl. 25, 1896	100,000
Brooklyn,	Cramp & Sons,	Aug. 27, 1896	350,000

CASTNER & CURRAN,

General Sales Agents for the Celebrated

Pocahontas Bituminous Coal.

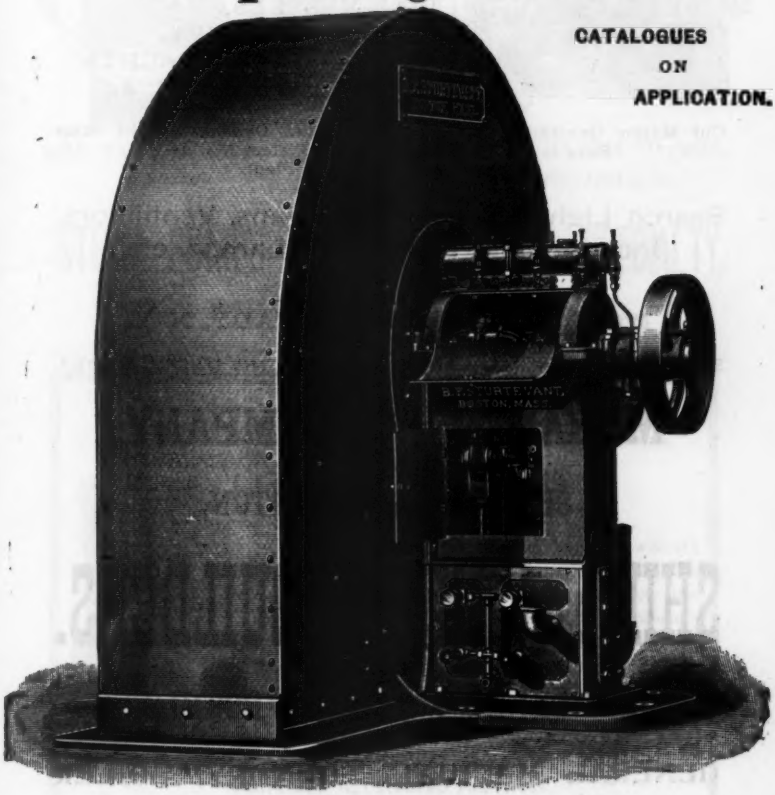
328 Chestnut Street, Philadelphia.
70 Kilby Street, Boston, Mass.
1 Broadway, New York.
Progress Building, Norfolk, Va.
Neave Building, Cincinnati, Ohio.
Terry Building, Roanoke, Va.
Old Colony Building, Chicago, Ill.
4 Fenchurch Avenue, London, Eng.

THE STURTEVANT

STEAM BLOWERS

FOR FORCED COMBUSTION.

CATALOGUES
ON
APPLICATION.

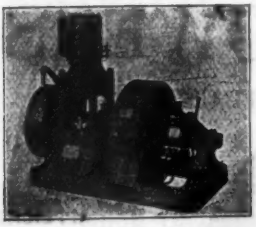


B. F. STURTEVANT CO.,

BOSTON, MASS.

91 Liberty St., New York. 16 S. Canal St., Chicago. 135 N. Third St., Phila.

GENERAL ELECTRIC COMPANY.



COMPLETE

Electric Light & Power Plants

FOR

WAR VESSELS,
STEAMSHIPS,
YACHTS,
DOCKS, WHARVES, &c.

Our Marine Generating Set with Engine and Dynamo on the same Base is the most Compact and Perfect Marine Electric Light and Power Plant.

Search Lights, Dock Hoists, Fans, Ventilators, Incandescent Lights, Arc Lamps, etc.

MAIN OFFICE, SCHENECTADY, N. Y.

SALES OFFICES IN ALL LARGE CITIES IN THE UNITED STATES.

MORAN BROS. COMPANY,

SEATTLE, WASHINGTON,

SHIP and ENGINE BUILDERS.

STEEL AND WOOD VESSELS.

GENERAL MACHINE, FOUNDRY, BOILER AND FORGE WORK.

MARINE RAILWAY, 1,000 TONS CAPACITY.

THE GEO. F. BLAKE M'F'G CO.,

BUILDERS OF EVERY VARIETY OF

91 Liberty Street,
NEW YORK.

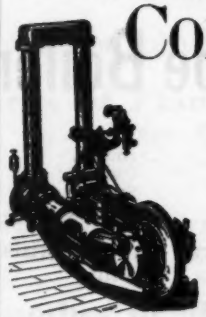
185 Devonshire St.,
BOSTON.

535 Arch Street,
PHILADELPHIA.

STEAM PUMPS FOR NAVAL USE A SPECIALTY.

Send for Illustrated Catalogue.

COMPOUND DUPLEX COMPRESSOR.



This Machine is intended for those who wish the most economical results, even in small plants. It is perfect in regulation, and as it can be run at very slow speed, unloading devices are unnecessary. Consumption of fuel and wear of machine are strictly proportioned to work done. Other attractive features.

ROCK DRILLS
AND MINING MACHINERY.

WRITE FOR ILLUSTRATED PAMPHLET.

RAND DRILL CO.

100 Broadway, - - NEW YORK.

SELDEN'S PATENT PACKINGS,

FOR STUFFING BOXES OF ENGINES, PUMPS, AIR COMPRESSORS, GLOBE VALVES, ETC.

THE "**SELDEN**" is in use in the U. S. Navy, and the largest Marine and Stationary plants in this and other countries, and its merits have been testified to by the repeated orders where it has once been introduced. The materials of which it is composed are entirely free from any substances which will either **score** or **corrode** rods and plungers, and will keep them tight with **less friction** than any packings on the market. A trial will convince you of the justice of these claims. It is put up in handy shape for the consumer and dealer.

Round, with Rubber Core, in sizes (varying by sixteenths) from ¼ to 2 inches diameter.
Round, with Canvas Core, in sizes (varying by sixteenths) from ¼ to 1¾ inches diameter.
Square, with either Rubber or Canvas Core, in sizes (varying by sixteenths) from 5/16 to 2 inches diameter.

RANDOLPH BRANDT,

38 CORTLANDT STREET, NEW YORK.

THORNYCROFT

Patent Water Tube Boiler

IN USE IN THE FOLLOWING NAVIES:

U. S. America,	Brazilian,	Indian,
British,	Danish,	Italian,
Argentine,	French,	Spanish,
Austrian,	German,	Etc., etc.

Used in the New 30-Knot "Destoyers" for the British Navy, and in U. S. Torpedo Boats "Cushing" and "Ericsson."

THE PATENTEES ARE PREPARED TO GRANT LICENSES FOR THE MANUFACTURE OF THE BOILERS UNDER ROYALTY.

AUTOMATIC FEED WATER REGULATORS.

SOLE AGENTS FOR U. S. A.:

THORPE, PLATT & CO.,

97-103 Cedar Street, NEW YORK.

CRAMP'S SHIP YARD,

PHILADELPHIA, PA.

BATTLE SHIPS, CRUISERS, PASSENGER AND FREIGHT STEAMSHIPS, &C.

Steam Machinery of every description, including boilers and all equipment, Marine Engines of any desired power, Mining Machinery, Hydraulic Plants, both for pumping and for power, Furnace Blower Engines, Tank Works; in short, every device or appliance embraced in the domain of applied mechanics.

Basin Dry Dock and Marine Railway.

Parsons' Manganese Bronze and White Brass.

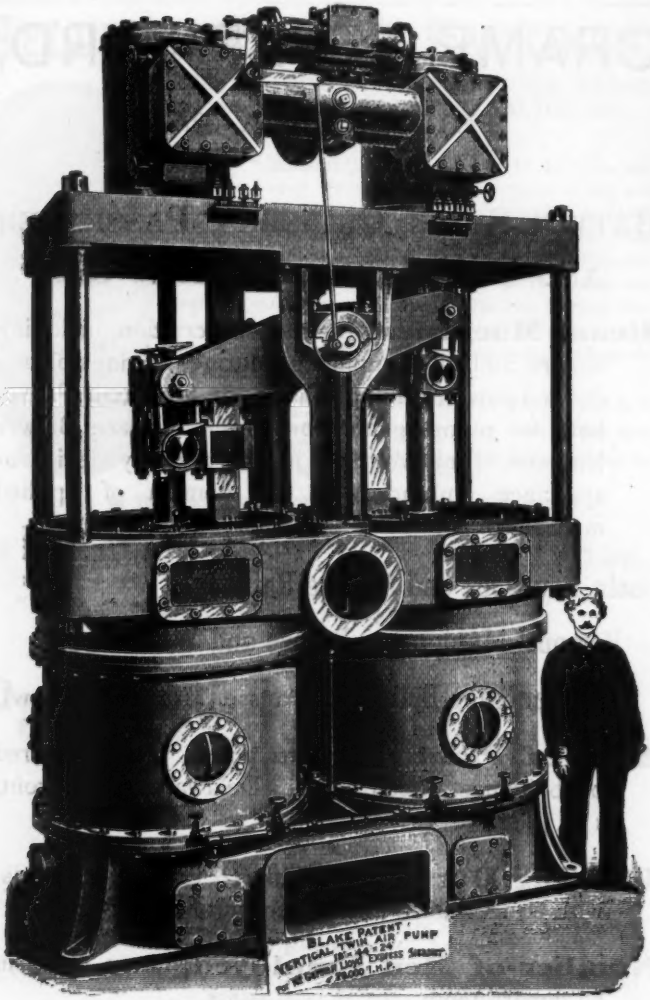
Water Tube Boilers (Niclausse, Mosher, Yarrow).

Area of Plant, thirty-two acres. Area covered by buildings, fifteen acres. Delaware River front, 1,543 feet.

Floating Derrick "Atlas;" capacity 130 tons, with 60 feet hoist, and 36 feet out-hang of boom.

Number of Men Employed, about 6,000 in all departments.

At Once, the greatest and most complete Ship and Engine Building establishment in the Western Hemisphere.



"THE LARGEST INDEPENDENT AIR PUMP IN THE WORLD."

ONE OF THE TWO

BLAKE VERTICAL "TWIN" AIR PUMPS

OF THE NORTH GERMAN LLOYD S. S. CO.'S STEAMER
"KAISER WILHELM DER GROSSE."

The following U. S. Vessels are also furnished with Blake "Twin" Air Pumps, viz: "Columbia," "New York," "Minneapolis," "Iowa," "Brooklyn," "Massachusetts," "Indiana," "Maine," "Gresham," "Daniel Manning" and "Hugh McCulloch."

NOTICE.

Subscribers to the **General Index**, who have not yet paid, will confer a favor by remitting the cost of the same, $1.25, as no bills will be sent.

The remaining copies of the edition will be disposed of to members at $1.25 per copy.

Respectfully,

F. C. BIEG,

Secretary-Treasurer.

ERRATA, MAY JOURNAL.

Page 373 for the eleventh item, read

Cubic feet swept per minute by L.P. piston per I.H.P.....	5.405	5.24	5.22	5.69

Page 388, read

t equals $s - p/2$, instead of t equals $s - q/2$

and w_1 " s_1 " w_2 " s_1

In the cut on page 386, the auxiliary ports in the seat are omitted from the development of valve face and seat.